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Heating · Piping and Air Conditioning

April, 1930

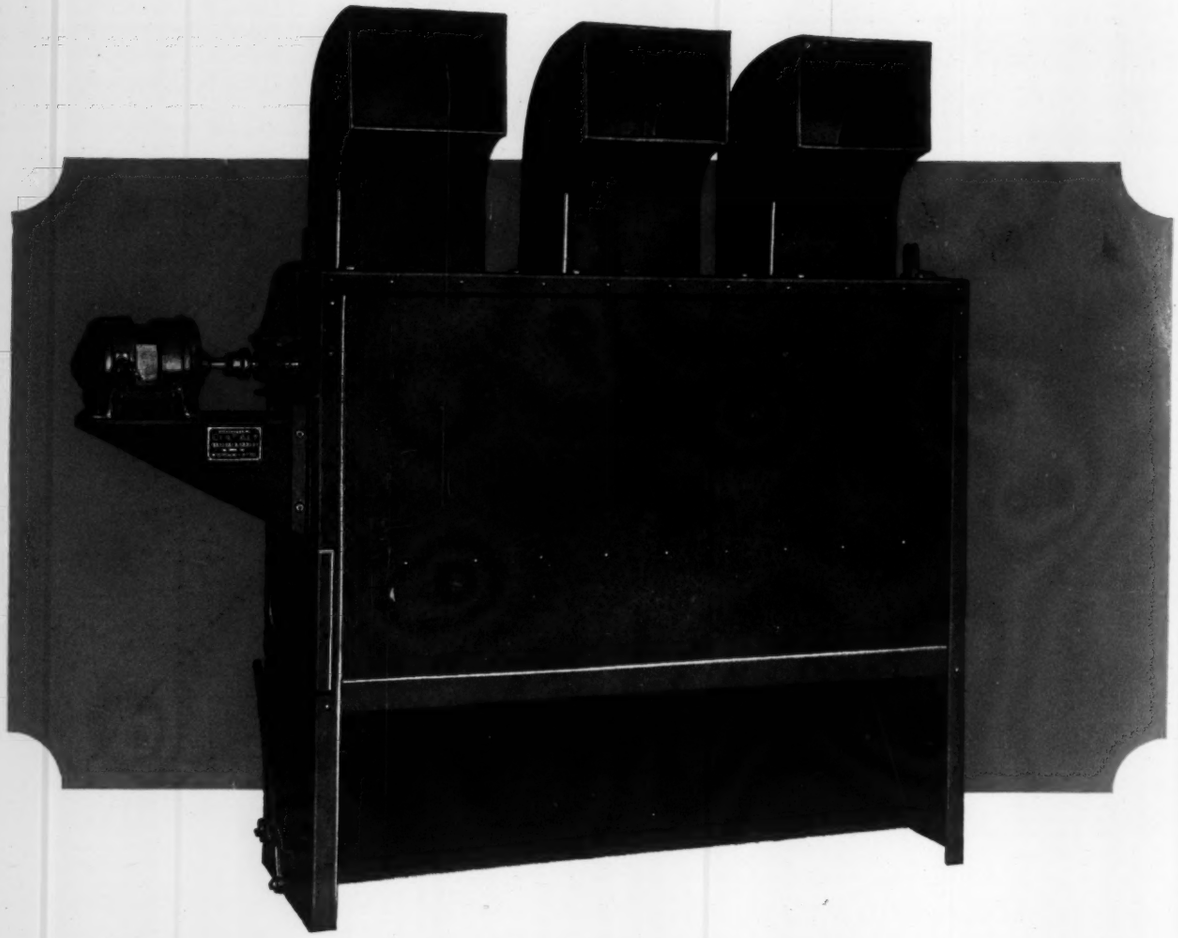
Articles on

Heating—
By W. R. WOOLRICH *and*
LEO HOLDREDGE

Piping—
By T. H. REA *and*
R. C. DOREMUS

Air Conditioning—
By ESTEN BOLLING
and PERRY WEST

Journal Section
of A. S. H. V. E.



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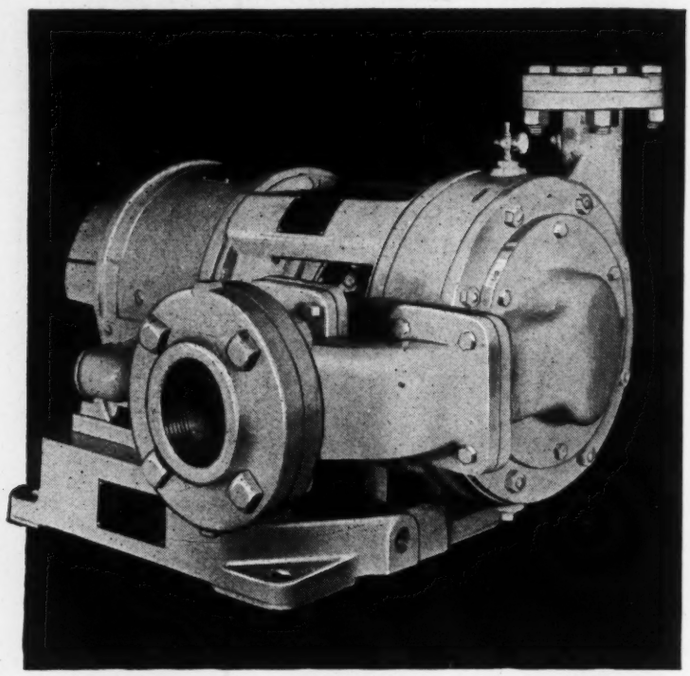

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Heating · Piping
and Air Conditioning

Vol. II April, 1930 No. 4

Whither Air Conditioning?

By Esten Bolling

No one knows, definitely, just what air conditioning will mean to our further industrial, public building and domestic development. Air conditioning is highly specialized, yet extraordinarily broad. As has been said so glibly so many times, air conditioning is fundamental to more than two hundred distinctly different industries, but this, unamplified, is not especially illuminating, is it?

Then let us examine a few indicative applications of air conditioning, with the thought that we may determine the present trends and, perhaps, inspire new trends as yet unthought.

THE last five years have seen a remarkable development of air conditioning applied for human comfort, rather than for industrial or production efficiency, which, for some ten years prior, constituted the principal rôle of air conditioning. The early industrial development was most logical and explicable, since the results of air conditioning, scientifically applied, were, at least to a sufficient extent, readily measurable in dollars and cents. That is, the early industrial applications were found to be definitely profitable.

On the other hand, the value of the first applications of air conditioning for human comfort and health, *sans* process or production considerations, was determined with extreme difficulty. What, for instance, was it worth, in monetary units, to maintain the comfort, vigor, alertness, *willingness, personal efficiency,* of a given group of employees at maximum, by means of air conditioning equipment, when that equipment served no determinable process or production function?

We know, knew then, many years ago, that this value is very great, but we were (and still are) unable to evaluate it definitely. Hence, those pioneering executives who first employed air conditioning for purely personal reasons; i.e., to protect the health and comfort of their employees, were pioneers indeed. They couldn't balance so much cost against so much gain. Yet many previsioning executives did adopt air conditioning for personal efficiency purposes solely when they discovered, in textile mills, for instance, that air conditioning equipment, installed to improve product and increase production, also had a significantly benign effect upon their absentee and labor-turnover records. These pioneers have enjoyed for years the *personal efficiency* benefits of air conditioning in addition to its obvious, calculable and extremely profitable process or production advantages.

Then, along came the movie theatre. "Summer," at first, meant "slump." Who'd go to a movie on a hot, humid summer night? Very few. So, in the early days of the movie the theatres were closed during the worst of the summer. Remember?

Now, however, with the aid of scientific air conditioning, the summer is the movie theatre's bonanza, because the conditioned—cooled, dehumidified ("It ain't the heat, it's the humidity")—theatre is a haven of refuge for the heat- and humidity-depressed citizen.

The movie theatre embraced air conditioning with fervor and dispatch. It was a life-saver. It made feasible the building of tremendous, costly theatres and obliterated the old practice of producing "summer pictures"—inferior, unsatisfying, but good enough for the small audiences which were willing to brave squalid, depressing atmospheres. Air conditioning made the movie theatre a year 'round institution. In some instances, in large cities such as New York, the summer profits repaid the whole cost of the air conditioning equipment in one or two seasons!

Here, at last, was an application of air conditioning for purely personal purposes which proved its value in explicit dollars and cents.

Also, and this is most important, the motion picture

theatre enabled air conditioning engineers, for the first time, to prove the *winter* value of air conditioning, a thing now surprisingly well understood by thousands of "lay" Americans. Summer "cooling" (really cooling plus *dehumidification,* the most important comfort and health factor) is spectacular and appreciated and much discussed, therefore. But the job the air conditioning equipment accomplishes in the movie theatre during the long winter is, though less obvious, of equal, if not greater, importance. It not only maintains a comfortable temperature but supplies the deficiency of atmospheric moisture which, ordinarily, makes artificially heated air drier than that which swirls above the Sahara, enervating, depressing, uncomfortable, unhealthful.

The unnumbered public is beginning to learn that air conditioning is as important during our long, cold, variable winters as during our very hot, humid summers. The motion picture theatre is helping it to learn that artificially heated air, lacking its proper percentage of moisture, is actually more uncomfortable than excessively hot air.

To shorten the story, then, the motion picture theatre, adopting air conditioning as a tremendously profitable investment, has become the means whereby the immeasurable advantages of this modern art are being made apparent to the general public, not only here but over all the world. There are conditioned theatres or public buildings nearly everywhere now—London, Paris, Delhi, Bombay—while here in the United States the halls of Congress in the Capitol and the White House offices are provided with this new-day magic which sets at naught both clime and season.

Whither, then, may air conditioning go?

Already, entire office buildings have been equipped to insure the comfort and guard the health of those who spend two-thirds of their waking hours within their walls. Not as a luxury—but because comfortable, healthy people are *efficient* people, producing more (and therefore more valuable) than enervated people, depressed by too-dry heated air or stiflingly hot, humid air.

Hotel restaurants and ballrooms have been equipped, that great gatherings of people, crowded into one space, may be comfortable, not only, but protected from the depressing, unhealthy conditions which invariably surround such assemblies in unconditioned buildings.

Today, in New York or Cincinnati, Los Angeles or Dallas, for instance, you may dine and dance, winter or summer, in comfort and in health, in conditioned hostelries where scientific air conditioning equipment banishes the heat or cold, the excessively humid or desert-dry atmospheric conditions which used to despoil every other attraction which ingenuity and money could afford.

Today, in Detroit, New York, Boston, Seattle, Fort Worth, Dallas, Philadelphia, Cincinnati, Indianapolis, Tulsa, you may shop comfortably, as well as economically and conveniently, in modern department stores where air conditioning not only insures *your* comfort and health, winter or summer, but the comfort and health of the salespeople who wait upon you.

The department store has discovered that air conditioning means more than even air conditioning engineers could, at first, conceive. Besides customer favoritism and employee efficiency, air conditioning has brought to the department store many other advantages. The cleansed, conditioned air avoids depreciation of displayed stock, through soiling. The hitherto barren summer season has been remade into an exceedingly profitable period. In the old days, before air conditioning, most department stores made their chief profits during the winter, school opening, Christmas holidays, Easter seasons. Summer often resulted in an actual loss. Today, with air conditioning, these stores have made the summer off-season one of plenty and profit.

Perhaps, here, we might digress, momentarily, to point out that these human comfort and health applications of air conditioning have been immeasurably advanced by mechanical refrigeration—machine-made cold.

Refrigeration equipment has been made available which is adaptable to the requirements of air conditioning, and remarkably effective in this service, and equally adaptable to many other services, from the freezing of ice to the cooling of rubber mill rolls. The advent of safe, efficient, automatic refrigeration has revolutionized the whole practice and potentiality of air conditioning. It has made air conditioning truly an every-day, year-'round facility, since it has made summer cooling and dehumidification quite as simple, economical and practically usable as winter heating and humidification. It has broadened the unnumbered applications of air conditioning for every conceivable process or personal purpose.

The near future will see, in my opinion, modern, scientific air conditioning applied not only to the restaurants and public rooms of great hotels, but to every guest room in the building! Think what this will mean. Comfortable, healthful rest and quiet, winter or summer. It will not be either necessary or desirable to open windows. The effective, smooth running air conditioning equipment will provide cleansed, conditioned air constantly and dependably, whether outdoors, blizzards howl or heat-waves undulate.

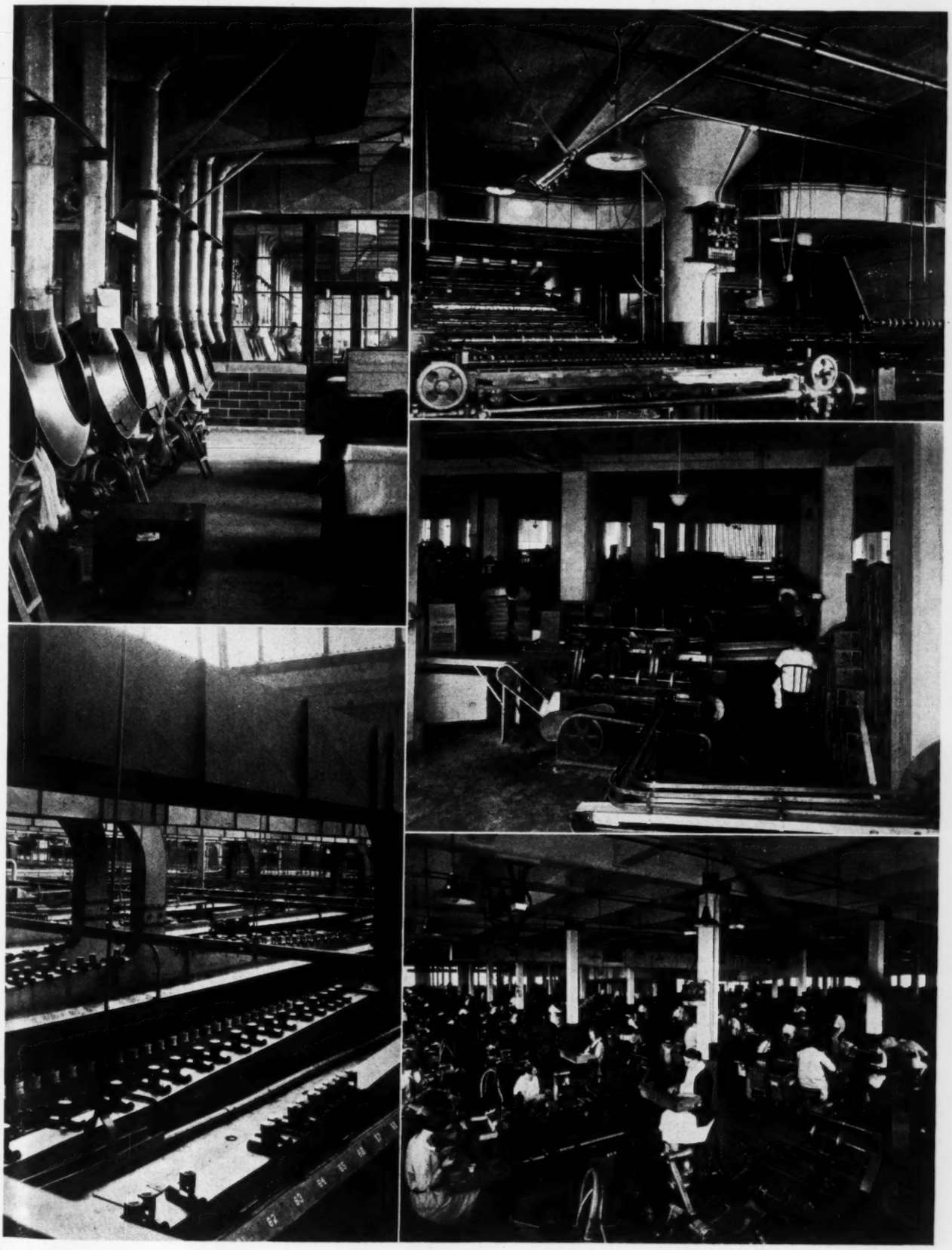
In hospitals the pain-wracked patient will be soothed and protected by mechanically perfected atmosphere. No drafts or dry, depressing heat, but uniform, health-giving, comfortable, conditioned air, draftless, cleansed, humidified and warmed, or dehumidified and cooled, as occasion may require. Contemplate this! It is unthinkable that it is not now a reality—that hospitals, those shelters of the sick and afflicted, do not provide the comfort and health of air conditioning as now available in any movie theatre worthy of the name!

Last summer one of my friends, an air conditioning engineer of note, was forced to enter a New York hospital for an appendectomy—in itself a simple, not especially dangerous excision. For days after we despaired of his very life—because "the weather"—hot, humid, devastating—imposed suffering which overcame people in perfect health, bowled over animals on the streets. This engineer has dedicated the rest of his life, and his rapidly increasing fortune, to the application of air conditioning's benign influence in the modern hospital.

Then, the office building. Several are now equipped. One, in San Antonio, was originally designed to incorporate scientific air conditioning throughout—every office, store, restaurant and corridor. Think what this means. Men and women spend two-thirds of their conscious lives in offices. Air conditioning not only provides the health and comfort of properly regulated temperature and humidity, of cleansed, invigorating air dis-

INDUSTRIALLY, AIR CONDITIONING WILL BECOME MORE INDISPENSABLE

CANDY (UPPER LEFT)—DUCT SYSTEM DELIVERS CONDITIONED AIR TO THE KETTLES. RAYON (LOWER LEFT)—CONDITIONED AIR IN A SPINNING ROOM ISSUES FROM THE PEPPER BOX OUTLETS ALONG THE MACHINES. PRINTING (UPPER RIGHT)—A PRESS-ROOM SUPPLIED WITH CONDITIONED AIR. FOOD (MIDDLE RIGHT)—CONDITIONED AIR IN A WRAPPING AND PACKING DEPARTMENT. TOBACCO (LOWER RIGHT)—CONDITIONED FRESH WORK OR CIGAR MACHINE MAKING DEPARTMENT



tributed uniformly without drafts, but, by obviating the necessity of opening windows, shuts out *noise,* the greatest bugbear of modern urban life.

Indeed, I think, with many others, including the able Dr. Luckiesh of G. E.'s Nela Park, perhaps our most eminent authority on lighting, that the office and apartment buildings of the future will be devoid of windows, or possess but a very few, installed because of some extraordinary outlook. Air conditioning will provide an ideal indoor atmosphere, immensely superior to any outdoor or "natural" condition, and scientific electric lighting will afford not only perfect, eye-kind, glare-free lighting but, very soon now, all the health-giving actinic rays of sunlight at its best.

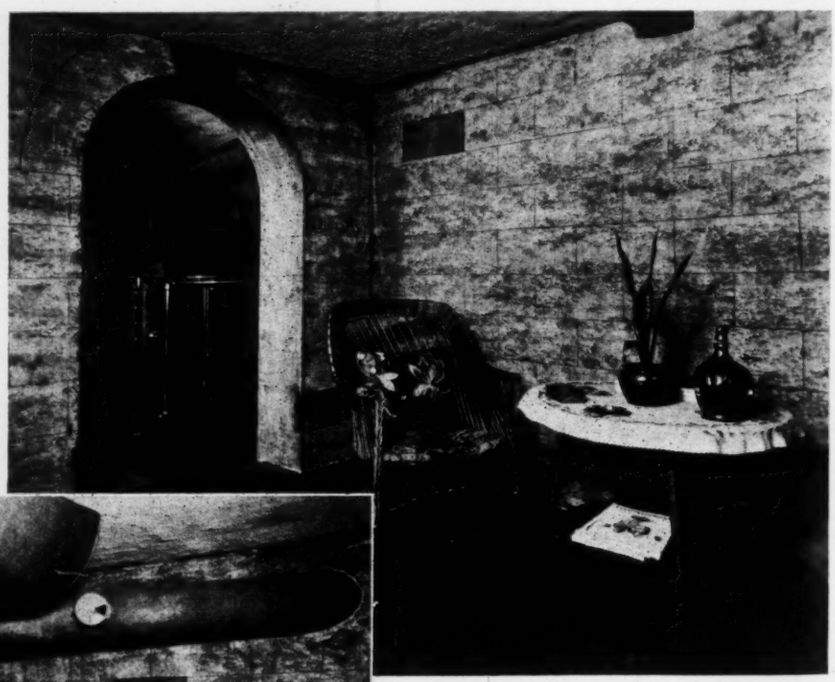
Bear in mind that sunlight is rarely, if ever, at its best—never in smoke-blanketed cities or within glassed enclosures. Thousands of people now utilize the modern sealed "sun-lamp," which converts electricity into those healthful actinic rays. It but remains for this lamp to be adapted to lighting as well and, lo, we have, indoors, no matter what the outdoor conditions may be (at least three-fourths of the time unfavorable in the northern United States), health-giving light rays very much more effective than those of Old Sol himself.

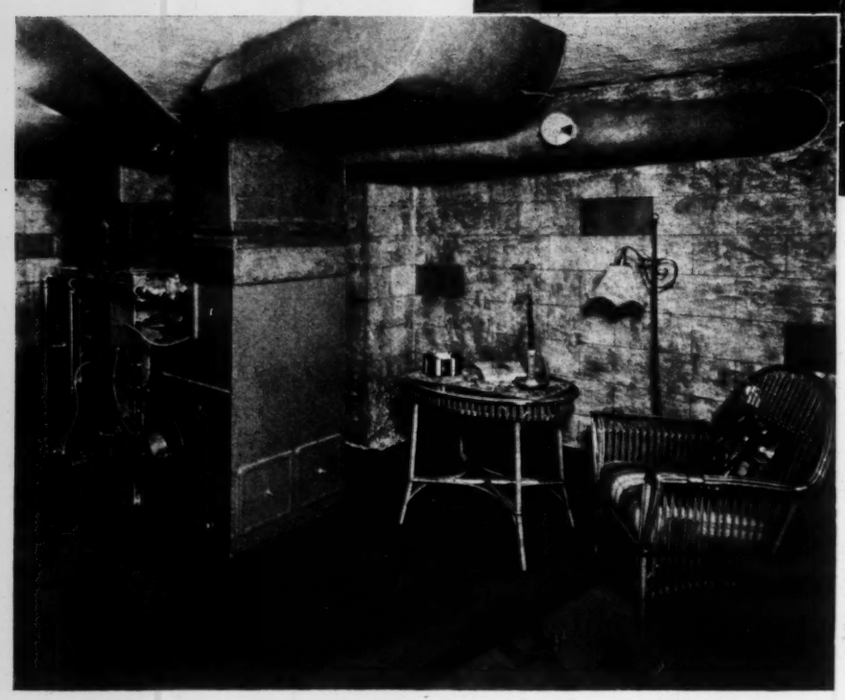
Then, there's the factory, the manufacturing plant of any description. In many industries air conditioning, from the process aspect alone, has wrought real miracles. In some industries air conditioning may fairly be credited with their very existence—rayon, for instance. This synthetic textile can not be commercially produced without the aid of scientific air conditioning, skillfully applied. This is a broad statement. I dare make it because I have heard it said by members of the world's greatest rayon-producing organization.

The present almost unbelievable production of machine-made cigars (superior to "hand-rolled" despite the ancient prejudice) could not have been achieved without air conditioning, simply because the delicately hygroscopic tobacco can not be machine-worked without positively accurate atmospheric control. And air conditioning has had much to do with modern methods of curing tobacco and preparing it for manufacture. Indeed, the flavor and "burning qualities," as well as the "color" of your favorite cigar, depend most probably upon the technical control afforded by air conditioning equipment.

This is true, also, of your best-liked cigarette. Air conditioning has assisted in the preparation and "ageing" of the various tobaccos and almost certainly governed the atmosphere within the rooms where the making machines have so fascinatingly tucked the blended tobaccos within their paper tube—with a rapidity and uniformity almost beyond comprehension. The "standard" cigarette making machine makes (not "rolls") more than 200,000 cigarettes per day, every one uniform and "free-smoking." Batteries of these machines make many millions of cigarettes every day—and make them efficiently, without constant attendance and adjustment, because air conditioning establishes and maintains the precise atmospheric conditions which keep the paper wrapper and the tobacco *always* in that condition which assures uniform, uninterrupted operation, whether it be bitter winter or stifling summer outdoors.

There are hundreds of similar examples where air conditioning has either made possible the development of a productive industry or assisted



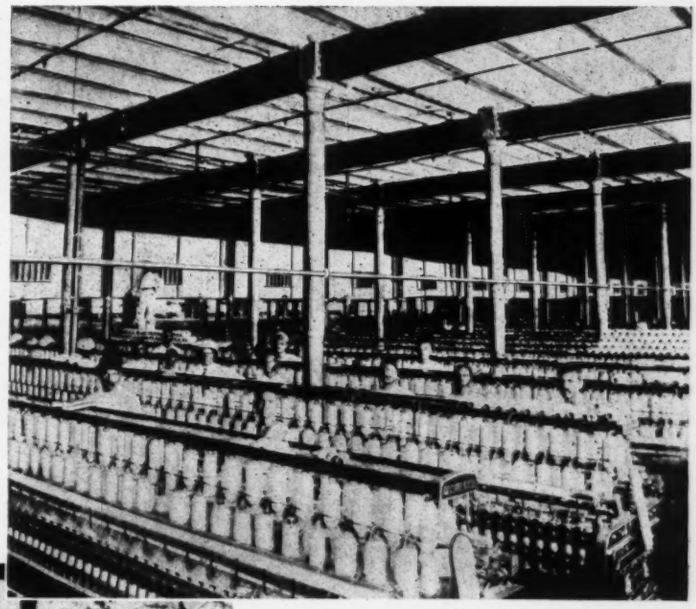


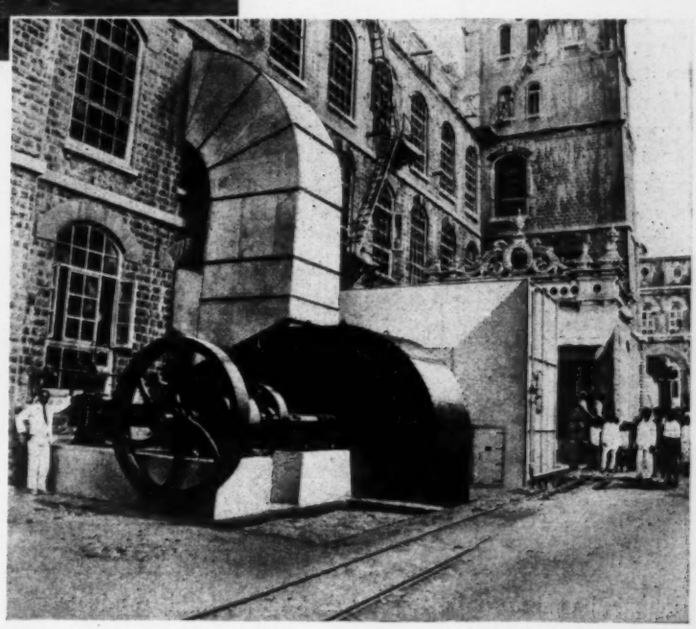
THE HOME
"IN ALMOST NO TIME NOW, AIR CONDITIONING WILL BE COMMON-PLACE IN HOMES." TWO VIEWS OF A GAS-FIRED AIR CONDITIONING UNIT IN A HOME

fundamentally in the industry's development. These examples range all the way from ceramics, sanitary ware—tubs, basins and wash-downs—to chewing gum, gun powder, corn flakes or airplanes.

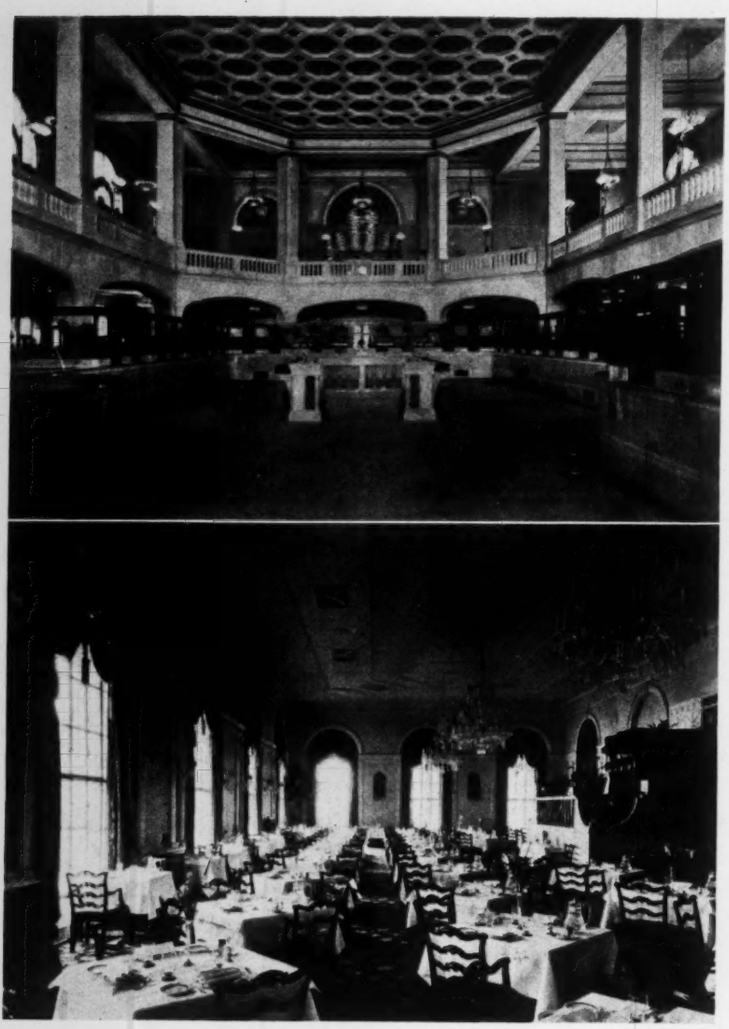
The facility of air conditioning surpasses our present-day imagination. It can accomplish or *assist* almost anything. It is essential in the quantity production of any product incorporating a hygroscopic material, or several hygroscopic; i.e., moisture- and/or temperature-sensitive materials. This includes wood furniture and all wood products; all textiles (cotton, wool, rayon, linen, in the order named), ceramic products (sanitary ware, brick, tile, pottery, abrasives, insulators, refractories, terra cotta, vitreous ware); confectionery (candies, chewing gum, chocolate); pharmaceuticals (coated pills, effervescent salts, capsules, infant or invalid foods, standardized medical products); still and motion picture films and studios; lacquers and varnishes; leather and leather goods; linoleum; paper and paper products of many descriptions, from "fibre-ware" to "papier mache"; artificial pearls; printing and lithography (the first important air conditioning undertaking was for the control of color printing; i.e., multiple-registration printing, in the early years of the present century); rubber (tires, dipped goods, hard rubber products); soap (bar and flake); all manner of food products, including, very importantly, our daily bread, other bakery products, fresh and cured meats, vegetables, fruit and cereals.

Most of these applications of air conditioning have been mentioned and many described. But experience has indicated repeatedly that applications of air conditioning for one purpose reveal other possibilities and eventually profoundly affect the whole development of the industry. Air conditioning was installed in a cereal plant to protect the crispness and cleanliness of the product. Shortly it was discovered that the properly conditioned air eliminated hitherto vexing and costly troubles with the machines which wax paper-wrapped the packed carton. Not only this, but the capacity of the machines was nearly doubled. An installation in a pharmaceutical laboratory





AIR CONDITIONING OVER ALL THE WORLD
IN SHALIPUR, INDIA, IN THE NARSINGGIRJI MILLS—(TOP), SPINNING ROOM FLOOR, SHOWING THE DUCTS FOR CONDITIONED AIR. (CENTER), NATIVES FABRICATING SHEET METAL. (BOTTOM), COTTON MILL CONDITIONING IN A CLIMATE OF 110 DEGREES IN THE SHADE



PUBLIC BUILDINGS PRESENT STEADILY INCREASING USES FOR AIR CONDITIONING
(BELOW), MAIN DINING ROOM IN A LARGE HOTEL. CONDITIONED AIR IS SUPPLIED THROUGH CLEVERLY CONCEALED OUTLETS IN THE CEILING. (ABOVE), LOBBY OF AN AIR CONDITIONED BANK

not only made it possible to pack the exceedingly hygroscopic product mechanically, but was found to improve the product so greatly that other installations were made for this initially unconsidered purpose.

An installation in a tobacco stemmery, made for the purpose of preventing leaf breakage due to excessive atmospheric dryness, not only accomplished this but coincidentally eliminated a most serious labor problem, by avoiding large turnover and absenteeism due to occupational disease caused by air filled with irritating dust.

In the manufacture of such odd things as artificial pearls, air conditioning, installed primarily for personal reasons, completely altered the whole manufacturing process, lowered production costs and vastly improved the product.

In countless industries and in many comfort and health applications, air conditioning has fundamentally affected the whole scheme of things—improving products in many ways, ofttimes unforeseen, and frequently leading on to further applications of great profit and efficiency.

In some cases, as with our textile industry, air conditioning has not only improved practice and product miraculously, but has actually changed the geographic center of the industry. A large part of our present textile industry now hums away efficiently and profitably in the very midst of the cotton fields from which spring its raw materials—made completely independent of unfavorable outdoor weather by means of air conditioning.

Thus, "whither air conditioning?" is a fascinating speculation, indeed.

First of all, there is every assurance that industrially air conditioning will become more and more indispensable and will find for itself ever new and important applications, revealing its own potentialities by actual accomplishments in service. In those industries where the product itself is not hygroscopic—these are few—air conditioning will be more widely applied for personal efficiency and, though proved profitable in this service alone, will, as heretofore, almost invariably discover to its users other benefits, other applications.

In that great realm of air conditioning for comfort and health we stand upon the threshhold—though thousands of eminently successful installations are now in operation. Refrigeration obviates all prior limitations and we shall see in the next few years astonishing strides in these applications of air conditioning.

Not only theatres but all types of public buildings will be conditioned, just as they are now merely "heated." Ships, freight and passenger railroad cars, even airplanes will be conditioned. In fact, far-seeing railroads have already adopted air conditioning and our de luxe trains, the finest in the world, will shortly provide additional comfort and cleanliness.

Then, in almost no time now, air conditioning will be a commonplace in homes, even the most modest. Hundreds of installations are even now in operation for winter conditioning and, tomorrow, highly-developed summer cooling equipment will be added. Developmental cooling apparatus has been in actual operation for several years and has proved entirely practicable. It is now but a matter of production.

Whither air conditioning? As we have said, no one knows. But assuredly a long way—industrially, in public buildings and in the home.

Committee on Properties of Metals Meets

The Joint Research Committee of the *A. S. M. E.* and *A. S. T. M.* on Effect of Temperature on the Properties of Metals held a meeting at the Book-Cadillac Hotel, Detroit, on March 18 in conjunction with the Spring Group Meeting of A. S. T. M. Committees. At this meeting, the committee decided to make a study of the high-temperature mechanical properties of cast and wrought carbon steels of customary composition. It is planned that these comparisons shall include the ordinary high-temperature tension tests and long-time or so-called creep tests.

It was also decided to study the high-temperature properties of one of the high nickel-chromium steels.

Ammonia Pipe Sizes

By R. C. Doremus

THE efficient operation of a refrigerating plant is primarily dependent on the sound judgment of the designing engineer in the wise choice of desirable equipment of proper proportions and capacities for each specified duty in the system. After the plant is installed, the operating engineer endeavors to operate the plant efficiently and usually meets with varying degrees of success, according to his own knowledge and capabilities, but he is always limited by what the designing engineer has made of the plant before he has taken charge of it. The complete system may be composed of several essential items of equipment, all of which are dependent on one another, for they are connected in a circuit of piping which conveys the refrigerant from one to another piece of equipment. The piping thus proves a most vital part of the equipment and it, also, must be wisely chosen.

Surface

The ammonia piping in the system may be partly used as evaporating surface or partly as mains and branches for conducting the refrigerant throughout the plant. When used as so much surface, the piping is usually 1¼-in. or 2-in. full weight steel pipe, the 1¼-in. being more prevalent in relatively small rooms, on account of the need of close centers and bends and low ratios and similarly the 2-in. being found more adaptable to larger rooms. One lineal foot of 1¼-in. pipe has an external surface of .434 sq. ft. or conventionally,

$$\text{Area equals} \frac{\text{Total lineal feet of 1¼-in. pipe}}{2.301}$$

One lineal foot of 2-in. pipe has an external surface of .622 sq. ft. or:

$$\text{Area equals} \frac{\text{Total lineal feet of 2-in. pipe}}{1.61}$$

From this it is evident that the ratio of 2-in. to 1¼-in. in area is $\frac{2.30}{1.61}$ or 2-in. pipe has 43 per cent more surface per lineal foot than 1¼-in. pipe. The ratio of the pipe alone in random merchant lengths is approximately in the same ratio as to cost, but the labor item and the possibility of using more lineal feet of 2-in. pipe per expansion in multiple and hence fewer necessary piping connections results in 2-in. pipe being cheaper per sq. ft. of surface for overall cost.

It is essential that the designer so plan the arrangement of pipe surface coils that the total amount of pipe per coil or per expansion valve does not exceed certain limits, otherwise excessive pressure drop and superheat of ammonia vapor will result. The customary maximum lengths of these vapor circuits vary with pipe sizes and are approximately as follows:

1-in.	900 lin. ft. max.
1¼-in.	1,400 lin. ft. max.
1½-in.	1,800 lin. ft. max.
2-in.	2,400 lin. ft. max.

These values may be exceeded by 10 per cent without any dire results, but are mentioned as a safe criterion to follow in general practice. The quantity of pipe or pipe ratios usually used as surface to maintain various temperatures in well-insulated cold storage rooms with 1¼-in. and 2-in. direct expansion piping coils was given in chart form on p. 3 in the January HEATING, PIPING AND AIR CONDITIONING.

Weight

Should the pipe installed be standard full weight, extra heavy or double extra heavy in weight? The majority of specifications will specify full weight for all surface coils, suction and low pressure vapor mains 2½ in. and larger and extra heavy weight for 2 in. and smaller to give structural strength with usual hanger spacing of 8 feet or less. Extra heavy weight pipe will be used throughout the remainder of the ammonia system main lines for all high pressure and miscellaneous piping connections sometimes subjected to high and sometimes to low pressure. Double extra heavy pipe is rarely used on an ammonia piping system because the pressures encountered do not warrant its cost. It may be found where other considerations, such as header constructions, welding, tapping, etc., require wall thickness. Lap weld will be provided in 2-in. and larger and butt weld in sizes smaller than 2-in. on account of preference and conventional mill rollings.

Pipe Sizes

The flow of the refrigerant through the piping conduit defines the size of pipe best suited for any given condition. In the case of liquid ammonia feed lines, the sizes are quite similar to those required for hot water carrying some steam, for the lines carry a volatile liquid containing more or less vapor. The velocity of flow in this case is of minor importance, but care is necessary in the arrangement of the piping layout to avoid gas pockets, which may impede the flow to a varying degree. Some installations containing gas pockets have been found to actually prevent the flow of liquid, due to the pressure of foul gas in the pocket.

The liquid drain line from the ammonia condenser to the receiver should be ample in size, free from traps and, if possible, should pitch to drain toward the receivers. This drains the liquid by gravity towards the receiver and allows any entrained vapor to return to the condenser to be condensed to liquid. Flow in this line in this direction is further induced by a vent or equalizer line, at least 1-in. in size and preferably larger in large systems, from the top of the receiver shell to the top of the condenser. In the unwise use of too small a liquid line or the absence of an equalizer pipe, the condensed liquid is liable to remain in the condenser, due to the receiver becoming "gas bound," which will cause a sud-

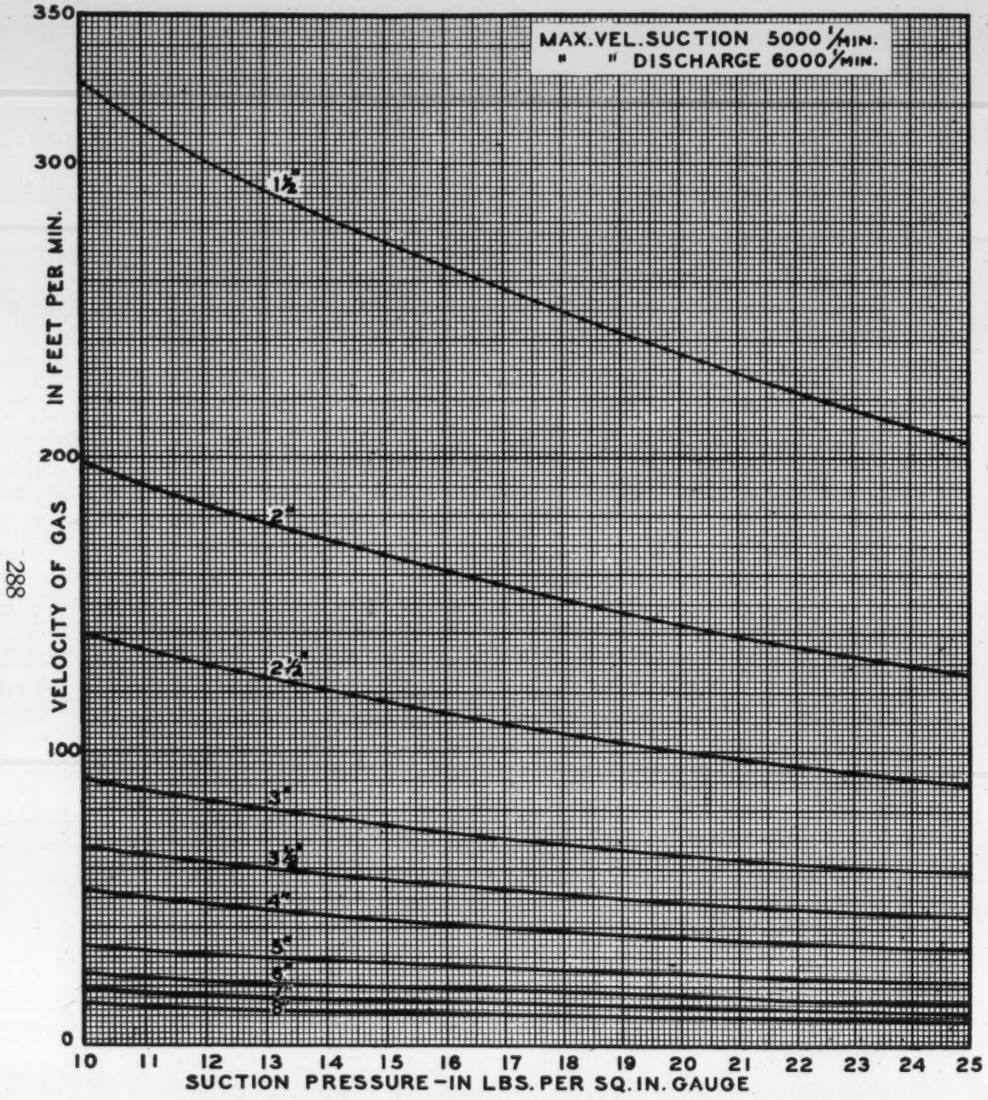


FIG. 1—VELOCITY CHART FOR AMMONIA AT A CONDENSER PRESSURE OF 155 LB. PER SQ. IN. GAUGE CARRYING ONE TON OF REFRIGERATION THROUGH PIPE SIZES FROM 1½ IN. TO 8 IN.

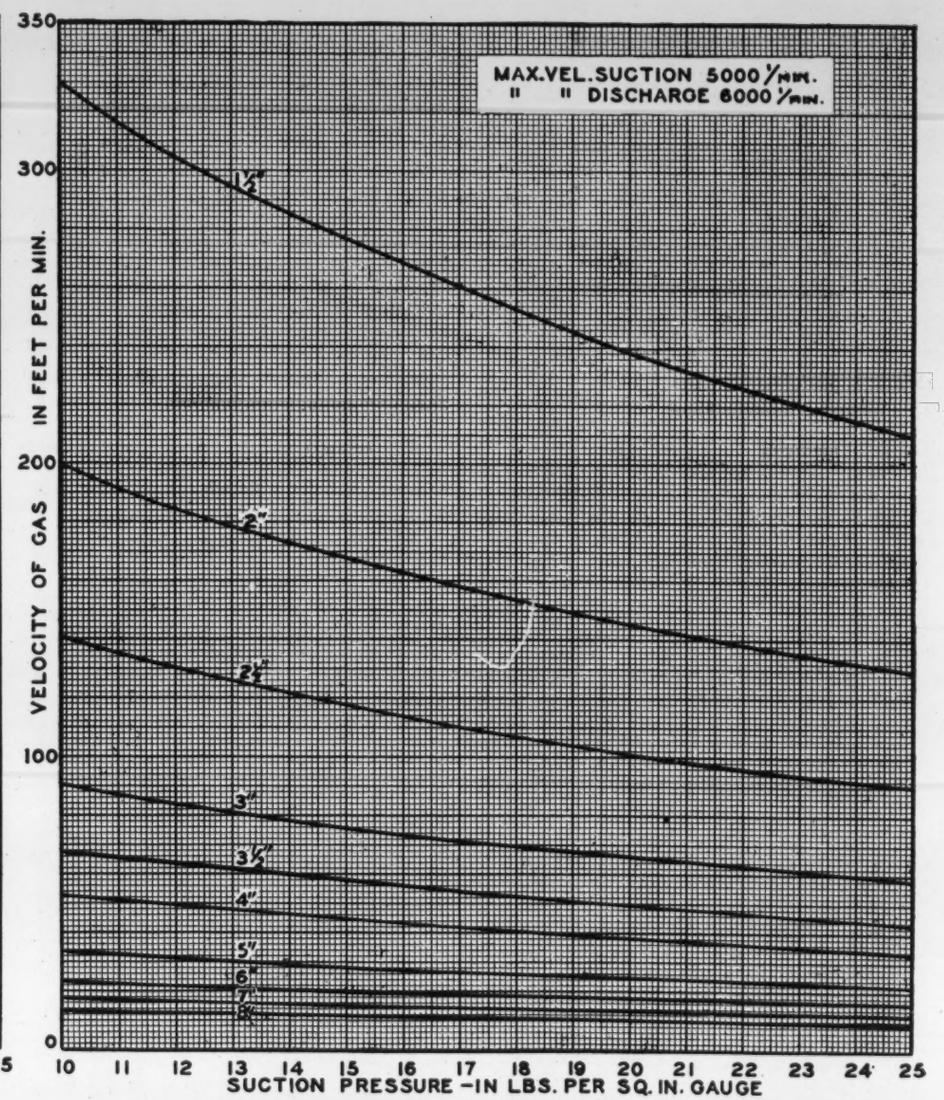


FIG. 2—VELOCITY CHART FOR AMMONIA AT A CONDENSER PRESSURE OF 170 LB. PER SQ. IN. GAUGE CARRYING ONE TON OF REFRIGERATION THROUGH PIPE SIZES FROM 1½ IN. TO 8 IN.

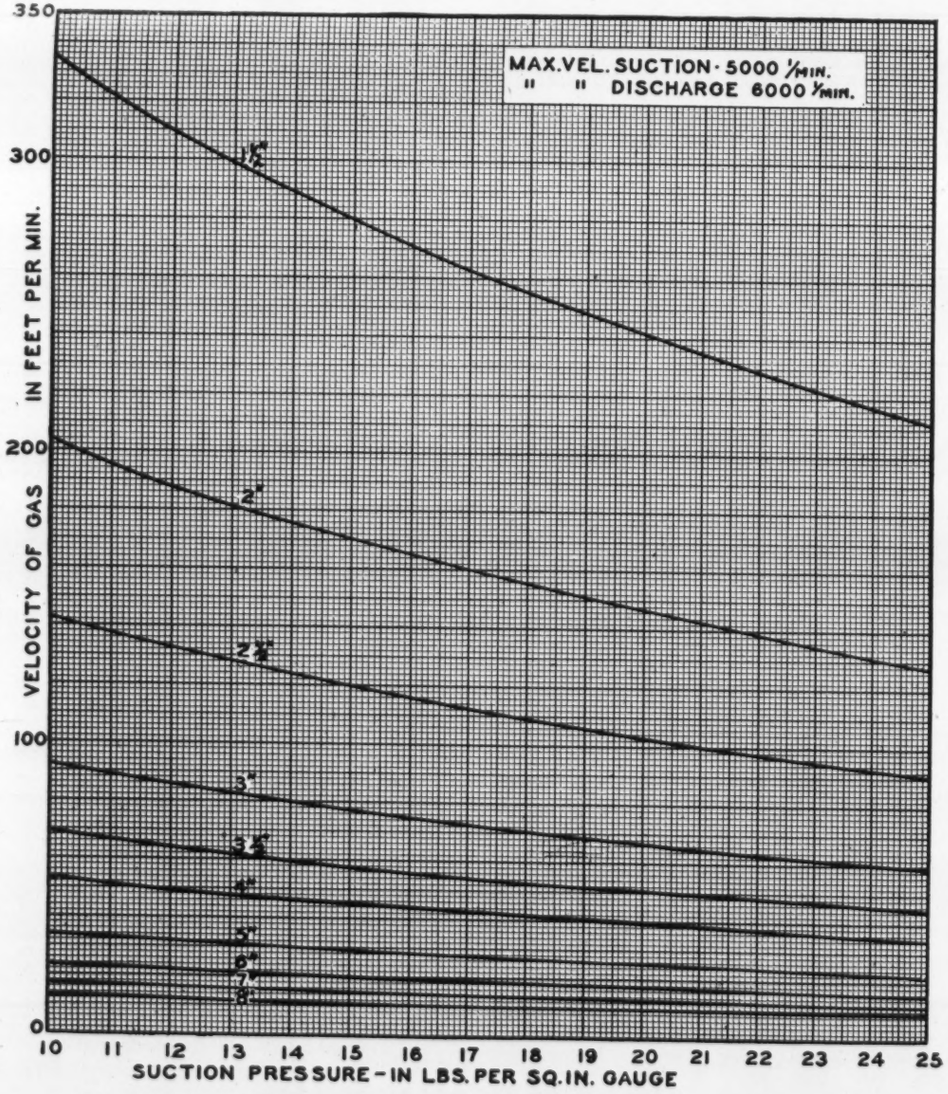


FIG. 3—VELOCITY CHART FOR AMMONIA AT A CONDENSER PRESSURE OF 185 LB. PER SQ. IN. GAUGE CARRYING ONE TON OF REFRIGERATION THROUGH PIPE SIZES FROM 1½ IN. TO 8 IN.

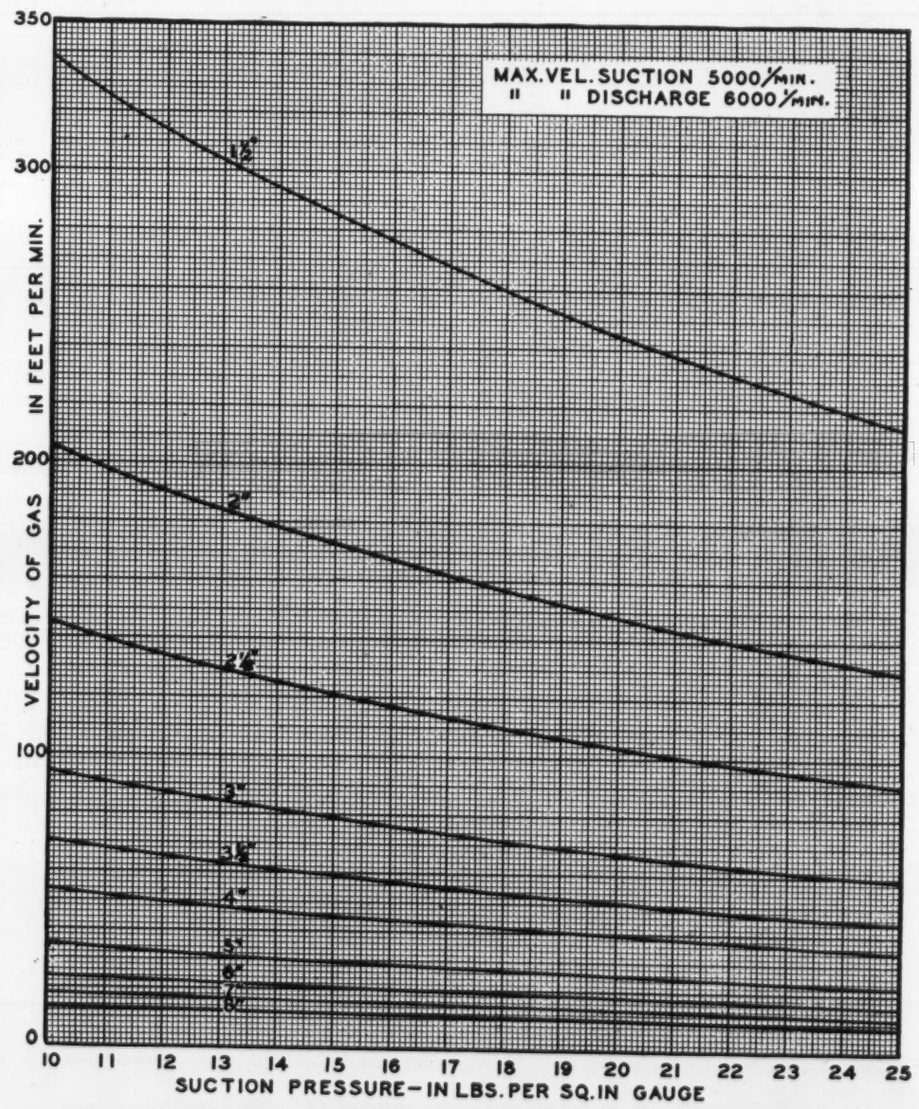


FIG. 4—VELOCITY CHART FOR AMMONIA AT A CONDENSER PRESSURE OF 200 LB. PER SQ. IN. GAUGE CARRYING ONE TON OF REFRIGERATION THROUGH PIPE SIZES FROM 1½ IN. TO 8 IN.

den increase in head pressure and may cause a break in the receiver liquid seal.

What size pipe will be most suitable for the discharge line? We may know we have a certain tonnage of refrigeration to convey through this line and we know in general that the vapor is the same quantity of vapor handled in the suction line. It is compressed by the compressor from the suction to the discharge pressure and if it remained at or near saturation it would be of much smaller volume. However, it is very much superheated, due to compression and to the heat equivalent of the internal work, and its specific volume is, therefore, high. This can be easily found by reference to ammonia superheat tables for any given suction and discharge pressure conditions, applying the theory of constant entropy during compression. It must be understood that cylinder wall heating, clearance and valve "wire drawing" will vary this slightly, but from a design standpoint these effects can usually be disregarded safely.

The determination of the most suitable size pipe line to use requires data on friction drops at various velocities under different conditions of superheat. If, for example, we consider a plant requiring an evaporator pressure of 15 lb. gage and the suction pipe is either excessively long or too small in size, causing an abnormally high velocity and a pressure drop of 10 lb. between the evaporator and compressor, it means that the ammonia compressor must produce a suction pressure of 5 lb. gage at the suction valve in order to produce that particular rate of flow. Similarly, if the discharge line is excessively long or too small in size, causing a large drop in pressure, the gage pressure at the compressor discharge valve may indicate 200 lb. per sq. in., whereas the actual condenser gage pressure may be only 185 lb. per sq. in. Either case, or a combination of both, means a lower actual vapor volume displacement from the compressor with a corresponding decrease in refrigeration tonnage capacity and greater hp. per ton required.

Therefore, it means that we must make an endeavor to decrease these pressure drop losses to an economic minimum and balance the cost of pipe line in capital investment against loss of capacity and cost of excessive power.

Some means of determining these frictional pressure drops in lines carrying superheated ammonia vapor have recently been undertaken by universities in connection with small pipe sizes, but there is little available information on drops in the larger sizes, with high velocities and various degrees of superheat. Future experiments will, no doubt, furnish such data. Observations have been made of actual operating conditions in commercial refrigerating plants from which it seems safe procedure to plan ammonia vapor pipe sizes in accordance with the following maximum velocities:

¾ in.	suction lines	2,000	f.p.m. aver. vel. per ton
1¼ in.- 2 in.	suction lines	3,500	f.p.m. aver. vel. per ton
2½ in.- 4 in.	suction lines	4,500	f.p.m. aver. vel. per ton
4½ in.-12 in.	suction lines	5,000	f.p.m. aver. vel. per ton
¾ in.- 1 in.	discharge lines	2,500	f.p.m. aver. vel. per ton
1¼ in.- 2 in.	discharge lines	4,000	f.p.m. aver. vel. per ton
2½ in.- 4 in.	discharge lines	5,000	f.p.m. aver. vel. per ton
4½ in.-12 in.	discharge lines	6,000	f.p.m. aver. vel. per ton

Installations in plants using these maximum limits have proven satisfactory and possibly higher velocities might be permissible, but these limits are safe to follow from a practical standpoint.

The user of refrigerating equipment may wish to buy an ammonia compressor with a standard rating of 50 tons capacity and the manufacturer may provide this machine with 3-in. inlet and outlet ports. Under one condition of pressure, a suction line of this same size as the port opening would be satisfactory, yet with a condition of lower back pressures and correspondingly higher volumes the velocities and drops would be excessively large if pipe were equalized with the port. So velocity charts are helpful in determining pipe sizes that will be suitable for all conditions.

It is obvious that these velocity conditions will vary with changes in suction and discharge pressures and with varying degrees of superheat, since these functions change the displacement of gas required for one ton of refrigeration. Also, it is generally agreed that the nearer to saturation it is possible to operate safely without priming, the less the losses will be in operation by avoiding re-expansion of liquid priming or the disadvantage of too high a superheat in the suction vapor.

Many engineers prefer a very slight superheat of approximately 5 deg. over saturation as being a safe operating condition. In order to establish a suitable constant and eliminate this function as a variable, a superheat of 5 deg. has been chosen as a desirable ideal. This leaves only two variables largely affecting the volume of vapor per minute per ton from which velocity charts shown in Figs. 1, 2, 3 and 4 have been prepared. These show the velocity of ammonia vapor in feet per minute through different pipe sizes to produce one ton of refrigeration at various suction and discharge pressures. For example, if we have an ice plant to consider with a suction pressure of 15 lb. gage and a discharge pressure of 185 lb. gage, the chart shows the velocity through the 4-in. suction line of one ton to be 45 ft. per minute. If we choose to use a maximum velocity limit of 4,500 f.p.m., the maximum allowable tonnage served by this line would be:

$$\frac{4500}{45} = 100 \text{ tons}$$

If our tonnage to be handled is actually 220 tons of refrigeration, by reference to the chart we find that with a 6-in. line the velocity will be:

$$20 \times 220 = 4{,}400 \text{ f.p.m.}$$

and with a 5-in. line

$$29 \times 220 = 6{,}280 \text{ f.p.m.}$$

which is too high, and we would therefore pick the 6-in. size as being most suitable.

In conclusion, it is obvious that good design of equipment and balance of the heat load may be discredited and operating efficiency spoiled by the arrangement of piping connections. Sweep pattern fittings should be used wherever possible to avoid close bends and sharp changes in direction of high velocity vapors avoided. Stream flow rules as applied to steam should be followed in a general way in piping design, although the theoretical formulae for ammonia and steam are slightly different. Sudden changes in velocity should be avoided inasmuch as they also cause pressure drops. In choosing between two sizes of pipe for any given flow, the larger should be used, giving a lower velocity if the case is questionable, for this will allow the least pressure drop and hence permit the higher operating efficiency.

Calculating Radiation Requirements

and Errors in "Rule-of-Thumb" Methods

By W. R. Woolrich and L. Holdredge

THE problem of computing the amount of radiation necessary for a given installation becomes complicated by the number of widely varying methods of computation proposed. There was a time when the amount of radiation was determined solely by the size of the room. Later, several other rule-of-thumb methods were developed. These rules may be used for estimates if sufficient allowances are made for unusual conditions.

The rule-of-thumb methods for calculating radiation requirements assume that the heat given off per square foot of radiation is constant. This may, in some cases, lead to a serious error. The actual amount of heat emitted per square foot from a radiator depends, of course, upon the height, number of columns and number of sections of the radiator, the room temperature and the steam pressure. More heat is emitted per square foot from a low radiator than from a tall one. There will be, also, more heat per square foot from a single column radiator than from one with two or more columns. The fewer the number of sections, the more heat will be given off per square foot. The greater the difference between the room temperature and the steam temperature, the more heat will be emitted per square foot of radiator surface.

Most approximate rules that have been offered make no allowances for variations in types of buildings, or locations. Such rules indicate no difference between the requirements of a brick house in Florida and a frame house of the same size in Montana. But quite evidently the amount of radiation actually needed is very different.

Heat Actually Emitted by Standard Types of Radiators

The amount of heat emitted per hour from standard sizes of radiators is given in Figs. 1 to 7. These plotted values are for a steam temperature of 215 F and a room temperature of 70 F. If the steam temperature is not 215 F the correction factors given in Fig. 15 must be used. Correction factors for room temperatures other than 70 F are given in Fig. 16.

Problem 1. How much heat will a 16-in., 6-section window radiator furnish if the steam pressure at the radiator is 6 lb. gage and the room temperature is 60 F?

Computation: From Fig. 7 the heat from the radiator per hour is 5450 B.tu under standard conditions; from Fig. 15 the correction factor for steam at 6 lb.

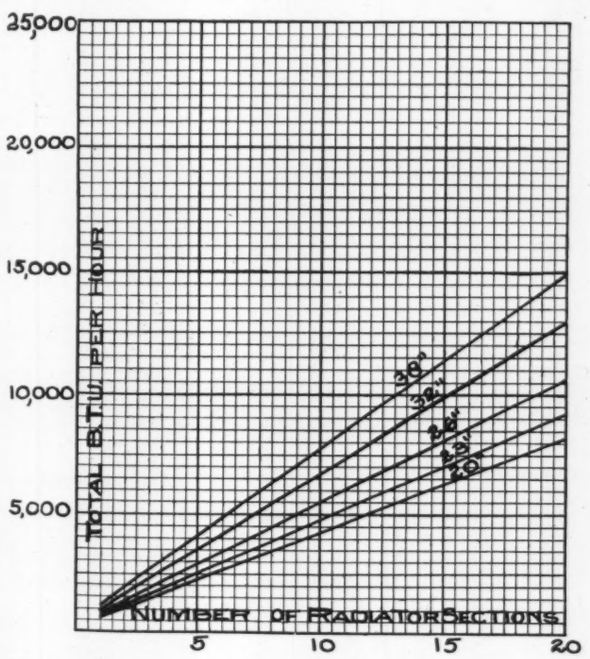


FIG. 1—HEAT EMITTED BY DIRECT RADIATION. SINGLE-COLUMN RADIATORS OF STANDARD HEIGHTS. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE 70 F.

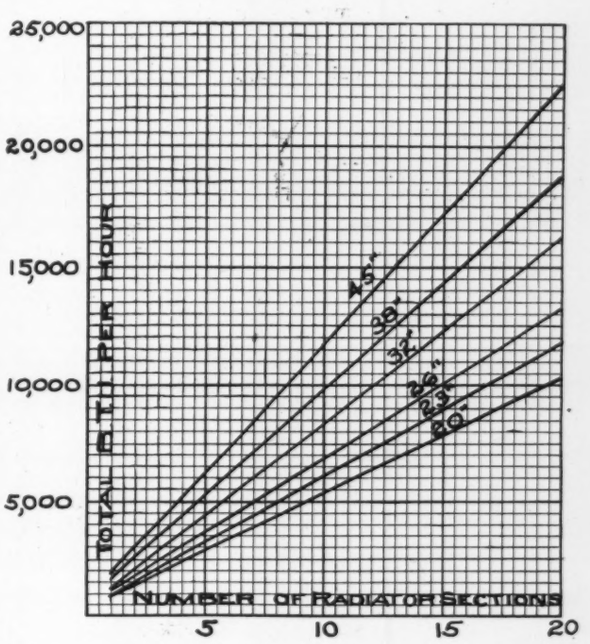


FIG. 2—HEAT EMITTED BY DIRECT RADIATION. TWO-COLUMN RADIATORS OF STANDARD HEIGHTS. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE, 70 F.

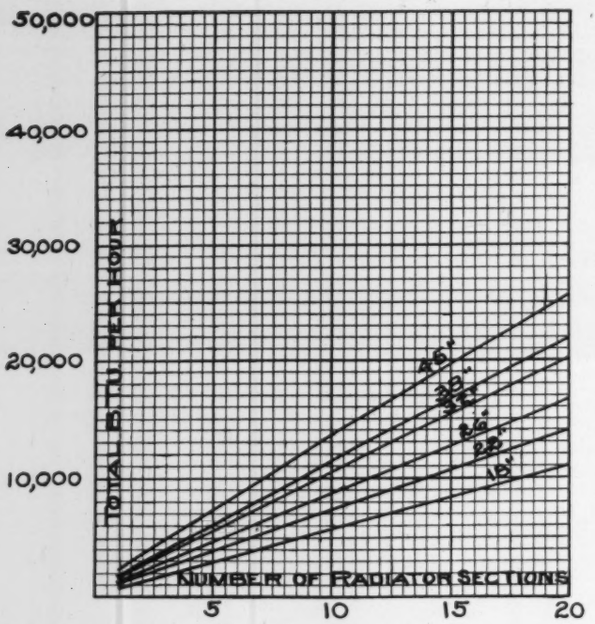


FIG. 3—HEAT EMITTED BY DIRECT RADIATION. THREE-COLUMN RADIATORS OF STANDARD HEIGHTS. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE, 70 F.

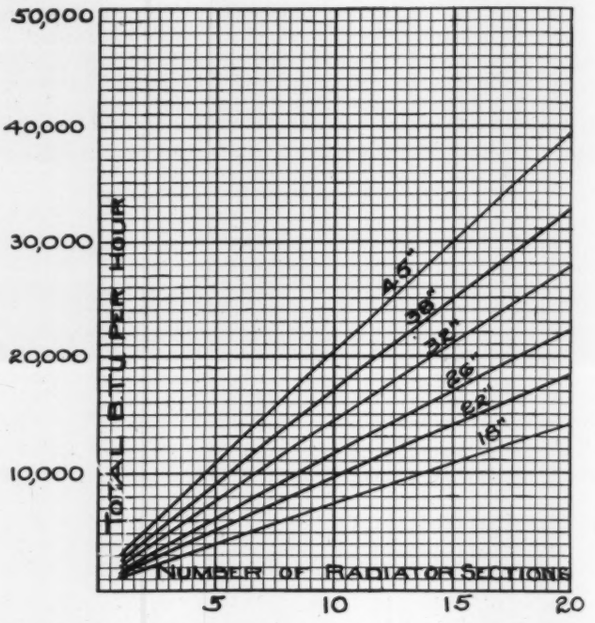


FIG. 4—HEAT EMITTED BY DIRECT RADIATION. FOUR-COLUMN RADIATORS OF STANDARD HEIGHTS. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE, 70 F.

gage pressure is 1.12; from Fig. 16 the correction factor for room temperature of 60 F is 1.06.

Then under these conditions the heat furnished by the radiator is $5450 \times 1.12 \times 1.06 = 6470$ B.tu *Answer.*

The heat emitted from standard sizes of radiators for hot water heating is given in Figs. 8 to 14. These figures are for a mean water temperature of 170 F and a room temperature of 70 F. This is a temperature difference of 100 F. For any other temperature difference, divide the heat emitted as given in Figs. 8 to 14 by 100, and multiply by the temperature difference.

Figs. 1 to 14 are based on data from the experiments of the *American Society of Heating and Ventilating Engineers* and the *Bureau of Mines.* The graphical presentation of these data has been found to be of considerable convenience in the selecting of radiator sizes.

Should pipe coils be used for radiators instead of the standard cast iron forms, Tables 2 and 3 will give the values of heat emitted per lineal foot of the three most commonly used pipe sizes when installed either as vertical wall coils or as ceiling radiators.

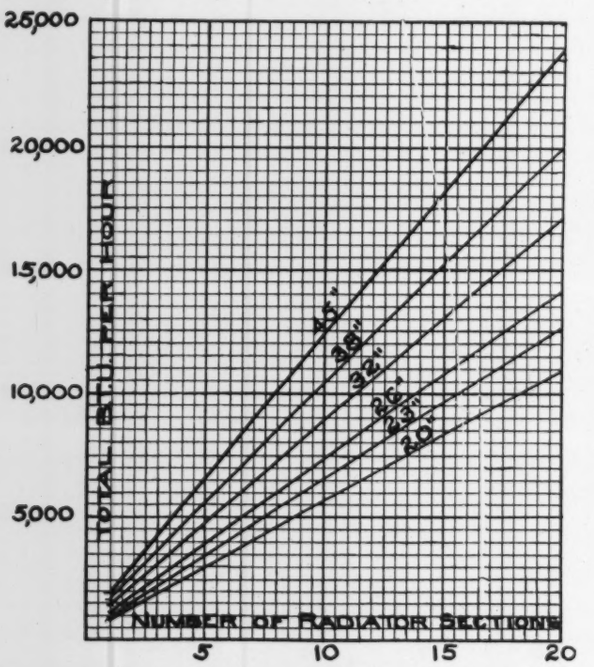


FIG. 5—HEAT EMITTED BY DIRECT RADIATION. TWO-COLUMN HOSPITAL RADIATORS OF STANDARD HEIGHTS. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE, 70 F.

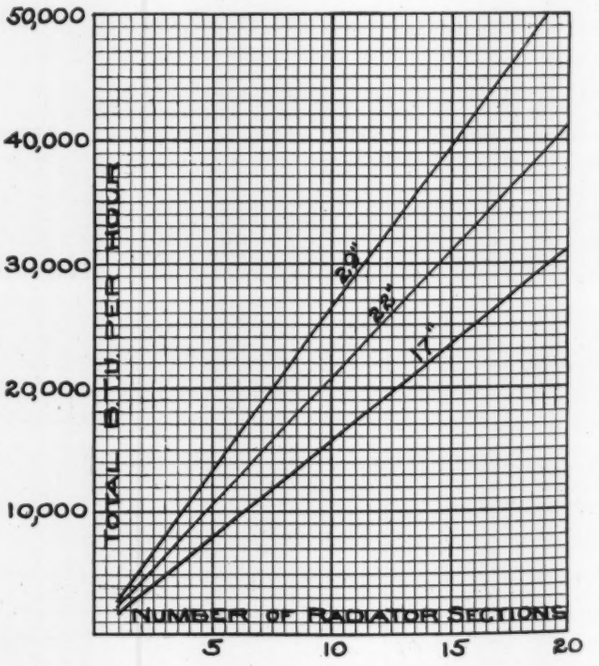


FIG. 6—HEAT EMITTED BY DIRECT RADIATION. WALL RADIATORS OF STANDARD SIZES. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE, 70 F.

TABLE 1—RATED SURFACE OF RADIATION FOR STANDARD SIZES OF RADIATORS. SQUARE FEET PER SECTION

KIND OF RADIATOR	HEIGHT											
	45	38	32	29	26	23	22	20	18	17	16	13
Single Column	...	3	2.5	...	2	1.66	...	1.5	.....	...	.....	...
Two Column	5	4	3.33	...	2.66	2.33	...	2	.....	...	.....	...
Three Column	6	5	4.5	...	3.75	.....	3	.....	2.25	...	.....	...
Four Column	10	8	6.5	...	5	.....	4	.....	3	...	.....	...
Two Col. Hospital	5	4	3.33	...	2.66	2.33	...	2	.....	...	.....	...
Wall Radiators	...	...	.....	9	.....	.....	7	.....	.....	5	.....	...
Window Radiators	...	...	.....	...	.....	.....	...	5	.....	...	3.75	3

TABLE 2—HEAT EMITTED BY PIPE COILS

STEAM TEMPERATURE 215 F ROOM TEMPERATURE 70 F
VERTICAL WALL COIL, PIPES HORIZONTAL

FORM OF RADIATION	B.TU PER LINEAL FOOT OF COIL PER HOUR		
	1 INCH	1¼ INCH	1½ INCH
One Pipe	137	168	192
Two Pipes	263	325	362
Three Pipes	362	441	504
Four Pipes	459	568	641
Six Pipes	590	731	826
Eight Pipes	679	835	942
Ten Pipes	762	942	1061
Twelve Pipes	842	1045	1178

TABLE 3—HEAT EMITTED BY CEILING COILS

STEAM TEMPERATURE 215 F ROOM TEMPERATURE 70 F

SIZE OF PIPE IN INCHES	B.TU PER LINEAL FT. PIPE	RADIATING SURFACE PER LINEAL FT. PIPE
1	123	.344 sq. ft.
1¼	151	.434 sq. ft.
1½	172	.497 sq. ft.

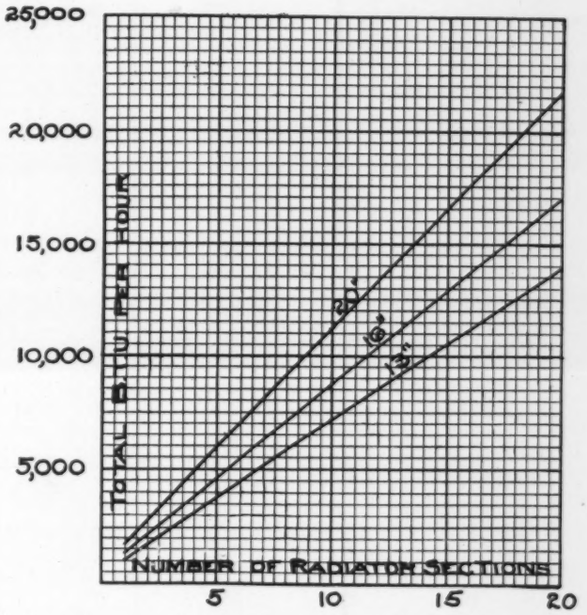


FIG. 7—HEAT EMITTED BY DIRECT RADIATION. WINDOW RADIATORS OF STANDARD HEIGHTS. STEAM TEMPERATURE, 215 F. ROOM TEMPERATURE, 70 F.

The amount of heat emitted from ceiling coils is less than from wall coils because the temperature is higher at the ceiling and circulation is restricted.

Comparison of the Several Rules for Finding Required Radiation with the Heat Balance Method By Typical Problem

To show the limitations of the approximate rules sometimes used, the following problems give an interesting comparison of results.

Problem No. 2. Select radiators for a room in Lincoln, Nebraska. It is to be continuously heated. The

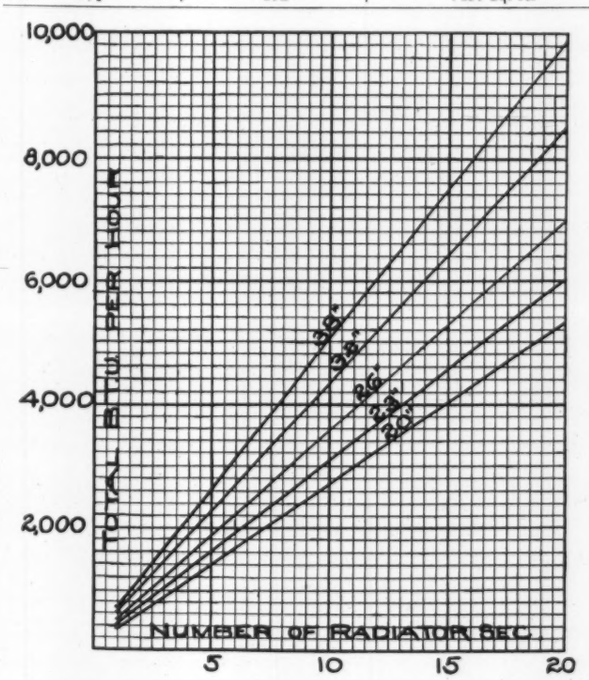


FIG. 8—HEAT EMITTED BY DIRECT RADIATION. HOT WATER, SINGLE-COLUMN RADIATORS OF STANDARD HEIGHTS. MEAN WATER TEMPERATURE, 170 F. ROOM TEMPERATURE, 70 F.

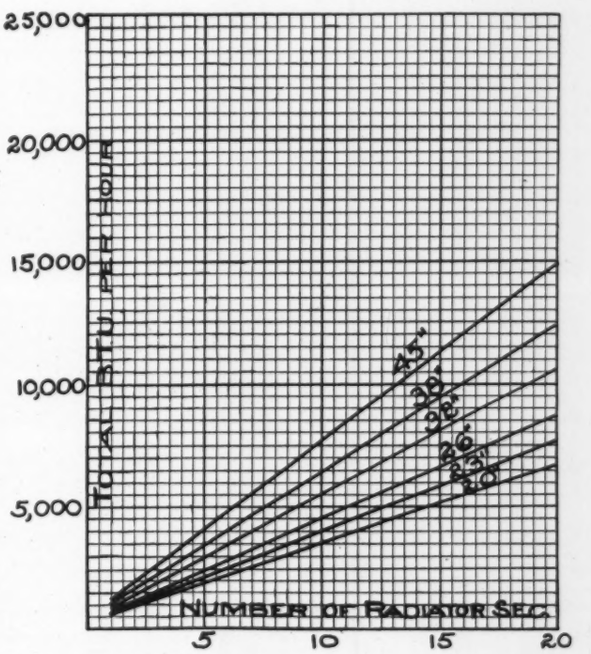


FIG. 9—HEAT EMITTED BY DIRECT RADIATION. HOT WATER, TWO-COLUMN RADIATORS OF STANDARD HEIGHTS. MEAN WATER TEMPERATURE, 170 F. ROOM TEMPERATURE, 70 F.

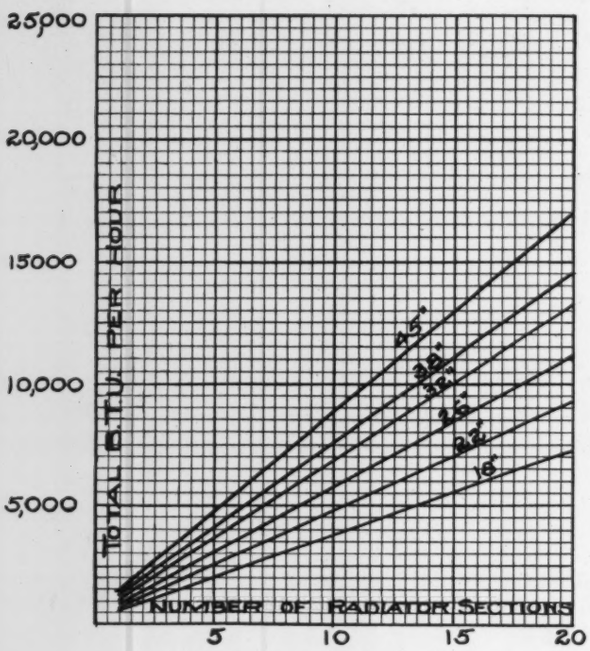


FIG. 10—HEAT EMITTED BY DIRECT RADIATION. HOT WATER, THREE-COLUMN RADIATORS OF STANDARD HEIGHTS. MEAN WATER TEMPERATURE, 170 F. ROOM TEMPERATURE, 70 F.

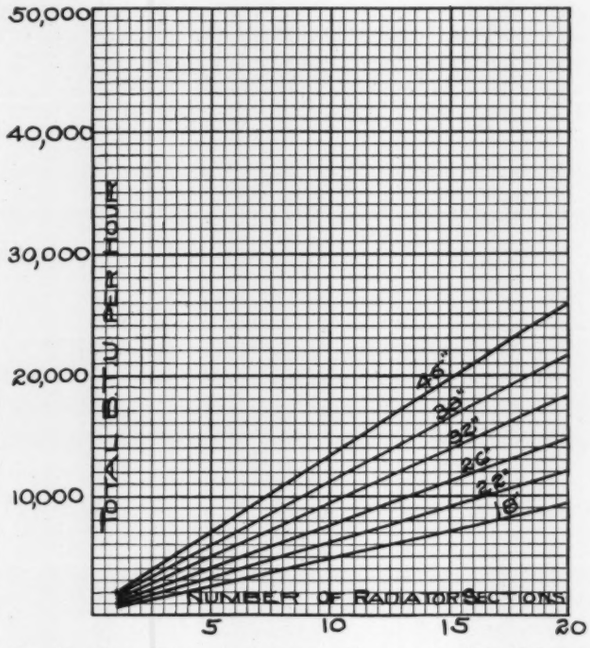


FIG. 11—HEAT EMITTED BY DIRECT RADIATION. HOT WATER, FOUR-COLUMN RADIATORS OF STANDARD HEIGHTS. MEAN WATER TEMPERATURE, 170 F. ROOM TEMPERATURE, 70 F.

height of the ceiling is ten feet. One exposed side is north and the other east. The north side is 30 ft. long and has 80 sq. ft. of windows and doors. The east side is 20 ft. long and has 50 sq. ft. window area. The exterior wall is 13-in. brick, plastered inside. The south and west walls are studs with lath and plaster on one side and are next to unheated rooms. The floor is double wood flooring over unheated basement. The ceiling is lathed and plastered with a finish plaster coat, built over joists with a single wood floor above. The room above is unheated. The room is to be heated to 70 F. Three changes of air per hour are required.

Computation:

1. By Mills' Rule

$$\frac{G}{2} + \frac{W}{20} + \frac{V}{200} = \text{radiation required in sq. ft.}$$

Glass area, $80 + 50 = 130$ sq. ft.

$130 \div 2 = 65$ sq. ft.

Exposed wall area, $(30 + 20)\ 10 - 130 = 370$

$370 \div 20 = 18.5$ sq. ft.

Volume, $30 \times 20 \times 10 = 6000$

$6000 \div 200 = 30$ sq. ft.

Total radiation by Mills' rule = **113.5** sq. ft.

2. By modified Mills' Rule

$$\frac{G}{2} + \frac{W}{20} + \frac{NV}{200} = \text{radiation required in sq. ft.}$$

Glass area, $80 + 50 = 130 \div 2 =$ 65 sq. ft.

Exposed wall area, $(30 + 20)\ 10 - 130 = 370$

$370 \div 20 = 18.5$ sq. ft.

Air supplied per hour, $30 \times 20 \times 10 \times 3 = 1800 \div 200 = 90$ sq. ft.

Total radiation by modified Mills' rule = 173.5 sq. ft.

3. By Baldwin's Rule

$$\frac{T_1 - T_0}{T_2 - T_1} A = \text{radiation required in sq. ft.}$$

The coldest recorded temperature of the Weather Bureau at Lincoln, Nebraska, is -29 F.

Difference between coldest outside temperature and room temperature,

$$70 - (-29) = 99$$

Difference between room temperature and steam temperature

$$215 - 70 = 145$$

and $$\frac{99}{145} = 0.683$$

Assuming 10 sq. ft. of exposed wall surface equivalent to 1 sq. ft. of glass surface,

$370 \div 10 = 37$ sq. ft.

$37 + 80 + 50 = 167$ sq. ft. of glass surface and equivalent wall surface.

$167 \times 0.683 =$ **114** sq. ft. of radiation required.

4. By Heat Balance Method

The design may be based on an outside temperature of 15 to 20 F above the lowest recorded temperature as given by the *United States Weather Bureau*. Since the Weather Bureau shows a minimum temperature of -29 F with a prevailing north wind, the design will be based on an outside temperature of -14 F. This makes a temperature difference between inside and outside of 84 F.

Computations:

(Data for heat required taken from tables in previous paper on "Heat Transfer Constants" in the February, 1930, issue.)

North wall:

Area $(30 \times 10) - 80 = 220$ sq. ft.

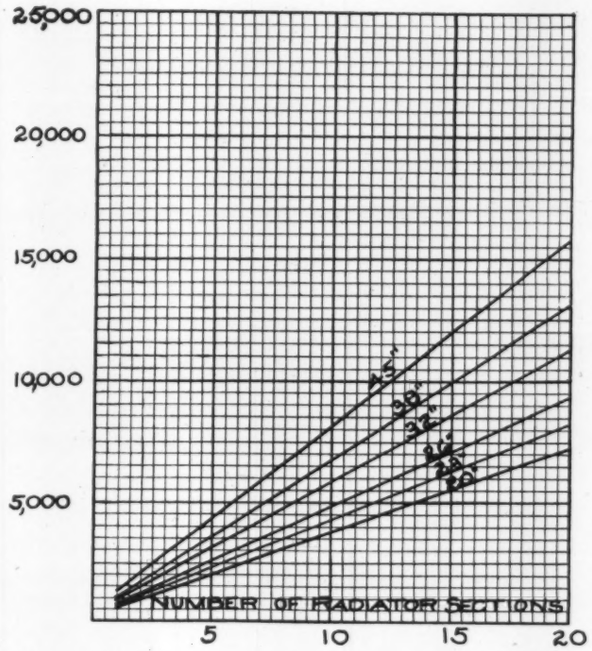


FIG. 12—HEAT EMITTED BY DIRECT RADIATION. HOT WATER, TWO-COLUMN HOSPITAL RADIATORS OF STANDARD HEIGHTS. MEAN WATER TEMPERATURE, 170 F. ROOM TEMPERATURE, 70 F.

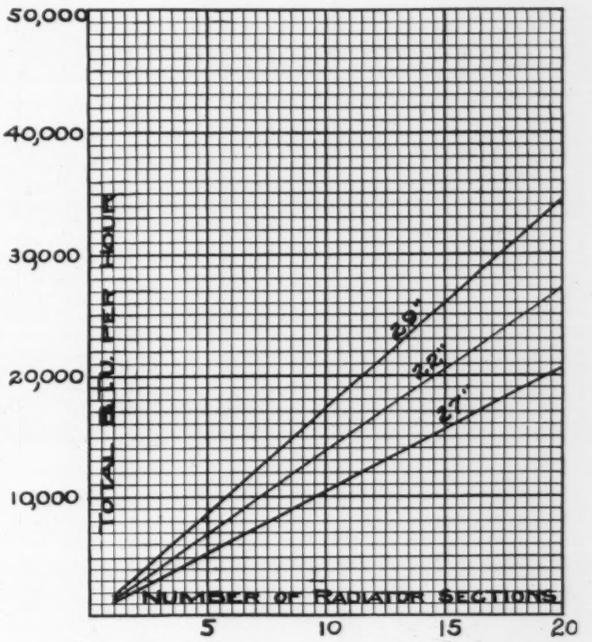


FIG. 13—HEAT EMITTED BY DIRECT RADIATION. HOT WATER, WALL RADIATORS OF STANDARD SIZES. MEAN WATER TEMPERATURE, 170 F. ROOM TEMPERATURE, 70 F.

Transmission constant for 70 F temperature difference = 17

For 84 F temperature difference this constant will be $\frac{84}{70} \times 22$

Add 10 per cent for northern exposure.

Heat loss will then be $1.10 \times \frac{84}{70} \times 17 \times 220 =$ **4937** B.tu per hour.

North wall windows:

Area of windows, 80 sq. ft.

Transmission constant for 70 F temperature difference = 78

For 84 F temperature difference this constant will be $\frac{84}{70} \times 78$

Add 10 per cent for northern exposure.

Heat loss will then be $1.10 \times \frac{84}{70} \times 78 \times 80 =$ **8236** B.tu per hour.

East wall:

Area = $(20 \times 10) - 50 = 150$ sq. ft.

Transmission constant for 70 F temperature difference = 17

For 84 F temperature difference this constant will be $\frac{84}{70} \times 17$

Heat loss will then be $\frac{84}{70} \times 17 \times 150 =$ **3060** B.tu per hour

East wall windows:

Area of windows = 50 sq. ft.

Transmission constant for 70 F temperature difference = 78

For 84 F temperature difference this constant will be $\frac{84}{70} \times 78$

Heat loss will then be $\frac{84}{70} \times 78 \times 50 =$ **4680** B.tu per hour.

South wall:

Area = $30 \times 10 = 300$ sq. ft.

Transmission constant is 21.

Heat loss will then be $300 \times 21 =$ **6300** B.tu per hour.

West wall:

Area = $20 \times 10 = 200$ sq. ft.

Transmission constant is 21.

Heat loss will then be $200 \times 21 =$ **4200** B.tu per hour.

Ceiling:

Area = $20 \times 30 = 600$ sq. ft.

Transmission constant 9.

Heat loss will then be $600 \times 9 =$ **5400** B.tu per hour.

Floor:

Area = $20 \times 30 = 600$ sq. ft.

Transmission constant 8.

Heat loss will then be $600 \times 8 =$ **4800** B.tu per hour.

Air changes:

Volume of room = $20 \times 30 \times 10 = 6000$ cu. ft.

Air to be heated 84 F = $6000 \times 3 = 18000$

Heat required to warm air to 70 F,

$$18000 \times \frac{84}{50} = 30240 \text{ B.tu per hr.}$$

Total = 71853 B.tu per hr.

Heat supplied by occupants:

Assume 10 persons doing clerical work.

Heat supplied by each person = 350 B.tu per hour.

Total heat from occupants = $350 \times 10 = 3500$

Total heat to be supplied = 68353

We are now ready to select the radiators. The room as described has six windows. A radiator should be placed under each window. Assuming that the radiation is equally divided we would need one-sixth of 68,353 or 11,392 B.tu per hour from each radiator. Next we need to know the height of the window sill above the floor, since the radiator should not come above the window sill. We may assume that a 32-in. radiator is the tallest it is possible to use.

Then we need to know what steam pressure is to be used. The heat emitted from radiators as shown in Figs. 1 to 7 is based on a steam temperature of 215 F. This corresponds to a gage pressure of approximately 1 lb. as shown by the lower curve in Fig. 15. If the selection of radiator is based on that temperature we may proceed with the data in Figs. 1 to 7 without the use of correction factors.

Then each radiator must supply 11,392 B.tu per hour and must not be over 32-in. high. Several different radiators will meet these conditions. They are as follows:

Type of Radiator	Height Inches	No. Sections	Heat Emitted B.tu per Hr.	Total Radiation (6 Radiators) Sq. Ft.
Two Column.....	32	14	11,436	280 (Fig. 2)
Two Column.....	26	17	11,299	272 (Fig. 2)
Three Column.....	32	11	11,360	297 (Fig. 3)
Three Column.....	22	16	11,323	288 (Fig. 3)
Four Column.....	26	10	11,435	300 (Fig. 4)
Two Column..... (Hospital)	32	13	11,341	260 (Fig. 5)
Two Column..... (Hospital)	26	16	11,378	256 (Fig. 5)
Two Column..... (Hospital)	23	18	11,327	252 (Fig. 5)

NOTE: There are several other radiators with a heat emission rating within 2 per cent of the desired amount. These might also be used.

Computations of Radiation Requirements for Room Temperature and Steam Pressure Other than Standard

If steam at a higher pressure is used, the amount of radiation should be decreased. Fig. 15 gives the correction factor to use with Figs. 1 to 7 when the steam is not at 215 F. This figure also shows the relation between steam pressure and temperature. Let us suppose steam at 10 lb. gage pressure is to be used. Fig. 15 shows that the temperature of steam at 10 lb. gage pressure is 240 F, and that the correction factor is 1.2.

A very convenient way to use this factor is to divide the total amount of heat which must be supplied to the room by the factor and then select radiation for that quantity of heat. This is evident from the mathematical analysis

$H_1 \times C = H_2$

Where

H_1 = amount of heat emitted as shown in Figs. 1 to 7

C = correction factor

H_2 = heat which must be supplied to room.

Then

$$H_1 = \frac{H_2}{C}$$

Applying this to the room described we have

$$H_1 = \frac{68353}{1.2} = 56961$$

We are now ready to proceed to select radiators as before.

When the room temperature is not 70 F, the correction factor to use is given in Fig. 16. Calculate the heat to be supplied to the room at the required temperature. Then use the correction factor from Fig. 16. This is done just as the correction factor for steam temperature other than 215 F is done.

If the room temperature is not 70 F and the steam temperature is not 215 F, both correction factors must be used.

Problem 3: Select four 2-column, 45-inch radiators of equal size to furnish 40,000 B.tu per hour. The room temperature is 60 F and the steam pressure is 15 lb.

Computations:

$H_1 \times C_1 \times C_2 = H_2$

Where

H_1 = heat emitted as shown in Figs. 1 to 7

C_1 = correction factor for room temperature

C_2 = correction factor for steam temperature

H_2 = heat which must be supplied to room.

$$H_1 = \frac{H_2}{C_1 \times C_2}$$

$C_1 = 1.058$ (Fig. 16)

$C_2 = 1.28$ (Fig. 15)

$$H_1 = \frac{40{,}000}{1.058 \times 128} = 29{,}537$$

$29537 \div 4 = 7384$

Now we select radiators for 7384 B.tu per hour from Fig. 2. The six-section, 45-inch, two-column radiator (Fig. 2) is the right size.

Comparison of Results of Computing Radiation Requirements for a Northern Location and a Southern Location

The "rule-of-thumb" methods gave far too little radiation for the room in Lincoln, Nebraska. This is largely because these methods are for average conditions in a climate not too cold. For warm climates they may give too much radiation. For comparison we might make similar computations for a southern city.

Problem 4: Select radiators for a room in Charleston, South Carolina, with the same construction and the same size as that given for Lincoln, Nebraska. This room is next to heated rooms on the south and west. The rooms

over it and under it are also heated. The room is to be heated to 68 F with steam pressure at 8 lb.

Computations:

By Mills' rule, (from Problem 2), radiation required = **113.5** sq. ft.

By Modified Mills' rule, (from Problem 2), radiation required = **173.5** sq. ft.

The coldest temperature recorded by the *United States Weather Bureau* at Charleston, S. C., is 7 F.

Difference between coldest outside temperature and room temperature, $68 - 7 = 61$ F.

Difference between room temperature and steam temperature is $235 - 68 = 167$ F.

$$\frac{61}{167} = 0.365$$

The glass surface and equivalent glass surface is 167 as in Problem 2.

By Baldwin's rule, radiation =

$167 \times 0.365 = \mathbf{61}$ sq. ft.

By Heat Balance Method:—As previously stated, the coldest temperature reached in this location, as shown by the Weather Bureau, is 7 F. The design might consistently be based on a temperature 15 or 20 F higher than the lowest recorded temperature. Since the prevailing wind during the heating season is from the north the design will be based on a 22 F temperature. This makes the temperature difference between the inside and outside $68 - 22 = 46$ F.

Computations:

(Data for heat required taken from tables in previous paper on "Heat Transfer Constants," published in the February, 1930, issue.)

North Wall:

Area = $(30 \times 10) - 80 = 220$ sq. ft.

Transmission constant for 70 F temperature difference = 17.

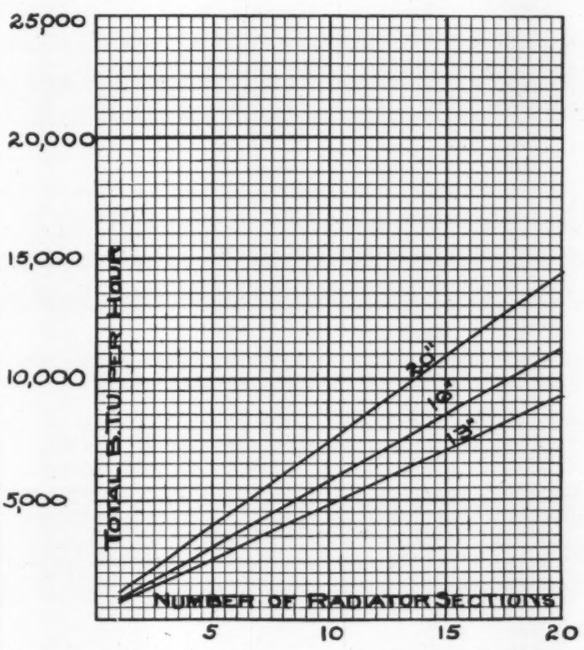


Fig. 14—Heat Emitted by Direct Radiation. Hot Water, Window Radiators of Standard Heights. Mean Water Temperature, 170 F. Room Temperature, 70 F.

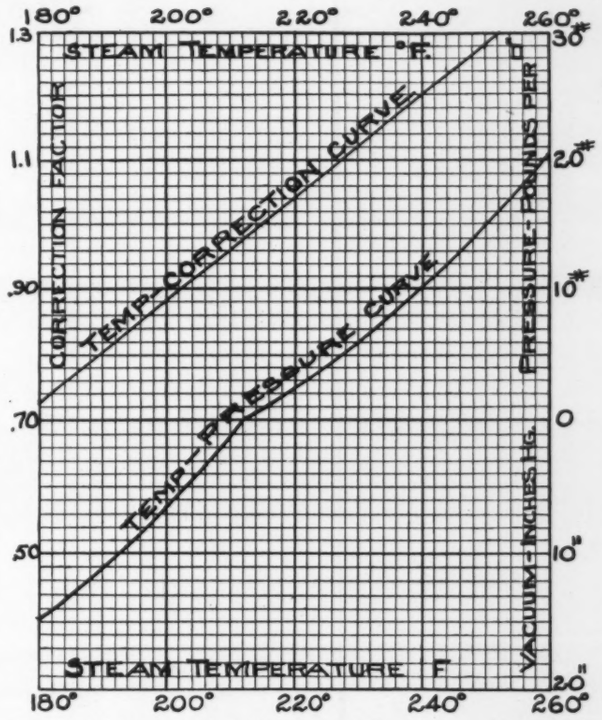


Fig. 15—Correction Factor for Heat Emitted When Steam Temperature is Varied from 215 F.

For 46 F temperature difference the constant will be $\frac{46}{70} \times 17$.

If we add 10 per cent for northern exposure then the heat loss will be $1.10 \times \frac{46}{70} \times 17 \times 220 =$ **2703** B.tu per hour.

North Wall Windows:

Area of windows, 80 sq. ft.

Transmission constant for 70 F temperature difference is 78.

For 46 F temperature difference this constant will be $\frac{46}{70} \times 78$

Add 10 per cent for northern exposure.

Heat loss will then be $1.10 \times \frac{46}{70} \times 78 \times 80 =$ **4511** B.tu per hour.

East Wall:

Area = $(20 \times 10) - 50 = 150$ sq. ft.

Transmission constant for 70 F temperature difference is 17

For 46 F temperature difference this constant will be $\frac{46}{70} \times 17$.

Heat loss will then be $\frac{46}{70} \times 17 \times 750 = \mathbf{1676}$ B.tu per hour.

East Wall Windows:

Area = 50 sq. ft.

Transmission constant for 70 F temperature difference is 78

For 46 F temperature difference this constant will be $\frac{46}{70} \times 78$.

Heat loss will then be $\frac{46}{70} \times 78 \times 50 =$ **2563** B.tu per hour.

South Wall—no heat loss.

West Wall—no heat loss.

Ceiling—no heat loss.

Floor—no heat loss.

Air changes

Volume of room, $20 \times 30 \times 10 = 6000$ cu. ft.

Air to be heated 46 F = $3 \times 6000 =$ 1800 cu. ft.

Heat required to warm air to 68 F is 18000 $\times \frac{46}{50} =$ **16560** B.tu per hour.

Adding all heat losses	B.tu per hour
North wall	2703
North wall windows	4511
East wall	1676
East wall windows	2563
Air changes	16560
Total =	**28013**
	B.tu per hour
Total heat loss	28015
Heat supplied by occupants (This as before)	3500
Total heat to be supplied	**24513**

Correction factor for 68 F temperature of room is 1.01 (Fig. 16)

Correction factor for 8 pounds steam pressure is 1.16 (Fig. 15)

$$H_1 = \frac{24513}{1.01 \times 1.16} = \mathbf{20924.}$$

For this problem it seems best to use only three radiators. If the radiators are all of the same size, each must emit one-third of 20924 B.tu or 6975 B.tu per hour under standard conditions.

If a single-column, 23-inch radiator be chosen, Fig. 1 shows that 15 sections will be required. Table 1 shows that this radiator has 1.66 square feet of radiating surface per section. This makes 25 square feet per radiator or 75 square feet for the room.

Hot Water Radiation Requirements

If hot water heating system is used, the radiator may be selected from the known amount of heat required by the use of Figs. 8 to 14. Problem 5 is given herewith, to show the method of computation.

Problem 5: Select a 38-inch, four-column, hot water radiator when 16500 B.tu per hour are needed under standard conditions.

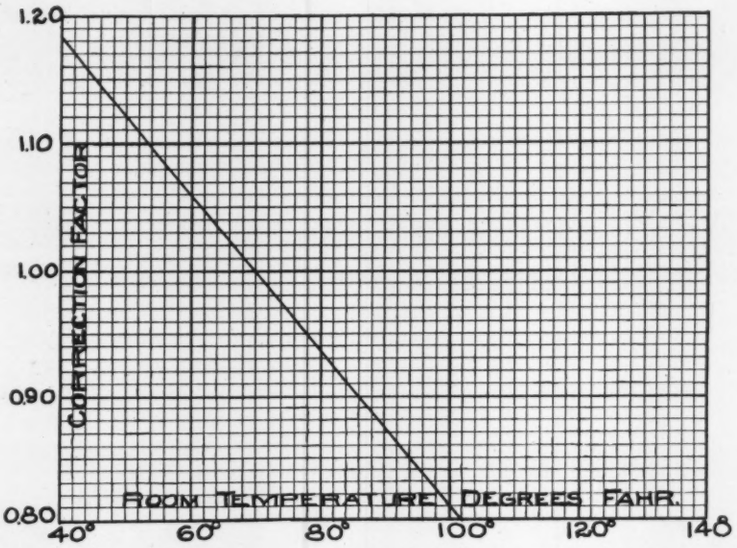


Fig. 16—Correction Factor for Heat Emitted When Room Temperature is Varied from 70 F.

Procedure: A 15-section, 38-inch, four-column radiator will supply the needed heat as shown in Fig. 11.

Problem 6: Select a 38-inch, four-column radiator to supply 16500 B.tu per hour with a room temperature of 68 F and a mean water temperature of 183 F.

Computation: Since Figs. 8 to 14 are for a room temperature of 70 F and mean water temperature of 170 F, a correction factor must be used. This correction factor is the temperature difference divided by 100. In this case it is $(183 - 68) \div 100 = 1.15$. Then, as before,

$$H_1 = \frac{H_2}{C} = \frac{16500}{1.15} = 14348$$

A 13-section, 38-inch, four-column radiator will give sufficient heat under these conditions.

While the two cases given here of heating in Lincoln, Nebraska, and Charleston, South Carolina, are not average conditions, they are far from the extreme. They show the danger in using the "rule-of-thumb" methods and the advantage of the heat balance method.

All the computations here are for direct radiation.

Refrigeration of Fruits and Vegetables

The research laboratory of the *National Canners Association* states that it is evident that an inert atmosphere does not in itself stop respiratory changes but alters their character. In many cases this altered respiration is detrimental, but there may be specific instances in which the results are beneficial. Adequate refrigeration checks respiration, be it in air or in an inert atmosphere. Temperatures several degrees below freezing stop all changes as long as the frozen state is maintained. Since freezing disrupts plant cells, it results in abnormal changes after thawing, which sometimes are undesirable. It also frequently affects physical qualities such as texture, although in many instances this can be avoided by the proper procedure.

The successful use of inert gases in the preservation of fresh fruits and vegetables must be in conjunction with refrigeration, and even with refrigeration the practical application of such a method is limited.

Mechanical Equipment

of the

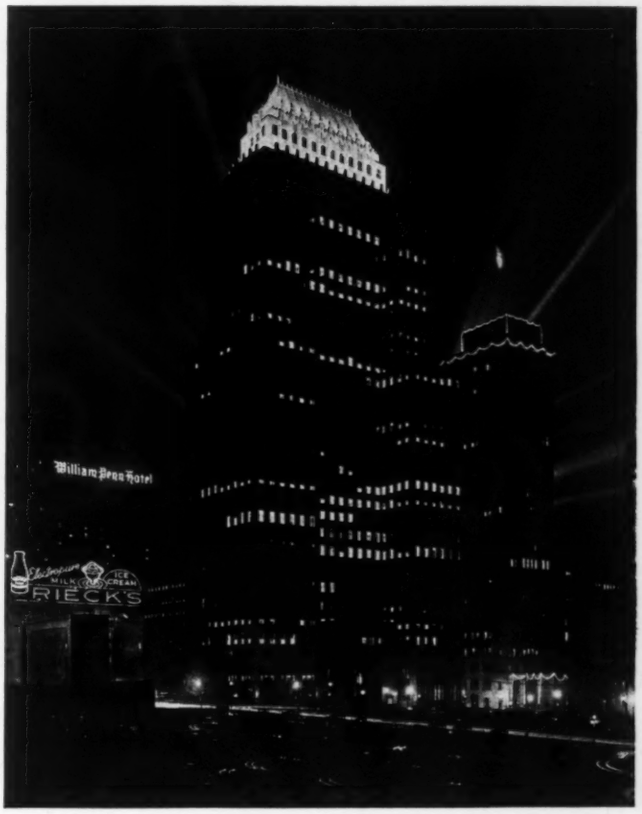
Koppers Building

By A. B. Whaley

A DESCRIPTION of the mechanical equipment and systems installed in the Koppers Building, in Pittsburgh, requires, first, a brief general description of the building itself:

The structure is thirty-four stories in height, has three floors below street level, and extends approximately 196 feet along Seventh Avenue, with a depth of 95 feet. Set-backs occur at the twenty-first, twenty-ninth, thirty-second, and thirty-fourth floors. The rentable area covers 260,000 sq. ft. and the contents total 6,462,000 cubic feet.

In the designing and construction of this modern office building, every effort was put forth by all concerned to erect a structure in which the mechanical, or commonly unseen features, would be consistent in every way with the high standard of the architectural treatment that was employed throughout.



A Combination of Three Heating Systems Was Decided Upon

Heating

Due to different requirements in various parts of the building, and taking into consideration first cost and operating expenses, a combination of three heating systems was decided upon, in which direct radiation, unit heaters, and indirect radiation serve definite areas.

It was early decided to purchase steam from a central heating station, and the necessity of automatic steam control became an important consideration.

After a primary and secondary pressure reduction in the heating company's lines, which enter the sub-basement on the William Penn Way side of the building, a third valve finally reduces the steam to its service pressure of one pound or less, depending upon the demand. From a 16-in. low pressure header, three separate branch headers feed the building risers, and two 8-in. lines extend to the sub-sub-basement for the fan heaters and water heaters.

Direct Radiation

The direct radiation system is two-pipe vacuum with up-feed the full height of the building, and totals slightly more than 51,000 square feet.

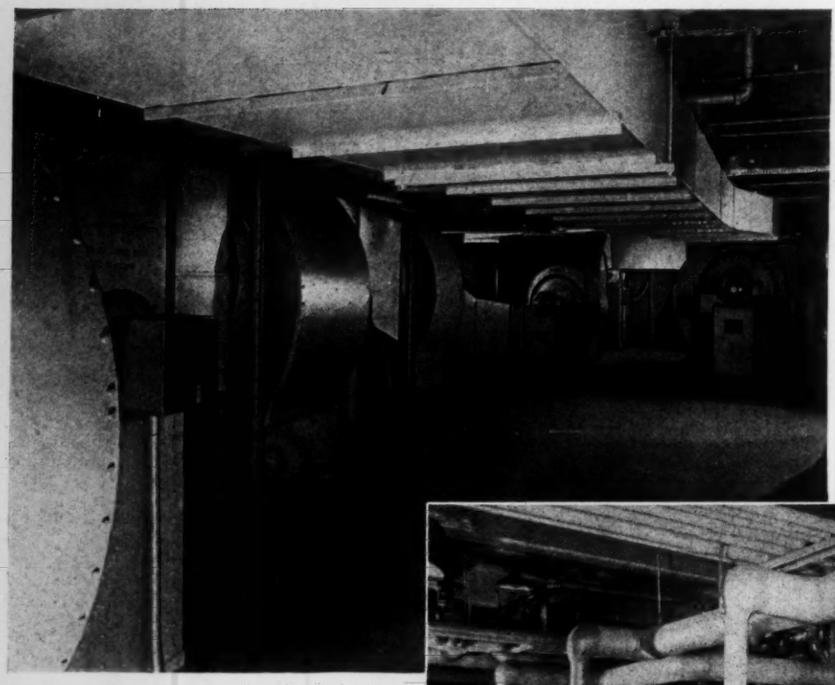
Thirty-three risers, all of which are located at exterior columns, are fed by three separate headers in the sub-basement, which divide the system into three main vertical zones, one along Seventh Avenue, and one at each end of the building. Further control over the system is provided by separate valves for each riser, so that should the necessity arise, small sections of the building may be heated independently of the whole.

Room temperatures, pre-determined to suit the tenant, are automatically maintained by means of a thermostatic supply valve on each radiator. With this provision it is felt that a long stride has been taken toward eliminating the wasteful practice of tempering air by opening and closing windows.

All radiators, with the exception of those in the elevator shaft and machinery rooms, are wall hung and totally enclosed in cork insulated recesses beneath the windows. The improved appearance and cleanliness of this arrangement, which was made possible by the structural design of the building, have more than offset the 10 per cent additional radiating surface that is required.

The return lines, of which there are 27 all located at exterior columns, run to the sub-sub-basement, where the condensate, after being metered, flows to a duplex automatic pump which discharges through the fan heater economizer coils to the heating company's return main.

As the heating system has not yet operated under its full normal load, no data are yet available by which to estimate costs. However, from the construction of the building with its 17-in. brick and stone walls, practically no openings in the rear wall, and automatic steam



ABOVE—EXHAUST FAN ROOM ON THE 33RD FLOOR

control on all radiators, we have every reason to believe that heating costs will be comparatively low.

Indirect Radiation

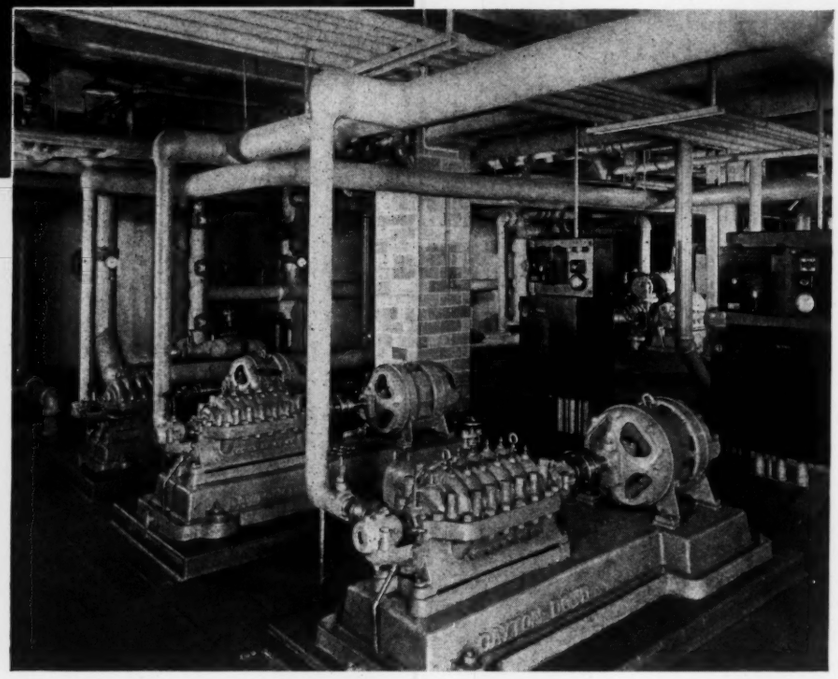
In conjunction with a mechanical exhaust system, the three floors below street level are all heated by indirect radiation only, and the first, second, and third floors, due to the nature of their occupancy and to the fact that the interior shops are considerably removed from radiators, are heated partly by direct and partly by indirect radiation. At the three entrances to the building, fresh air is discharged through radiators in which the steam supply is controlled by push buttons operating pneumatic valves, and indirect radiation also extends to the executive suite of the Koppers Company, on the fifteenth floor. This, however, was done more for the purpose of ventilation than heating.

The total heating surface of the steam coils and economizer coils by which the above seven floors are either partially or completely heated, amounts to approximately 3200 square feet, and 89,000 cubic feet of air are drawn through these coils per minute.

The temperature of the fresh air delivered by the fans is automatically controlled by pneumatic thermostats that operate diaphragm valves in the steam supply to the heating coils. For the sub-basement and sub-sub-basement, which are respectively storage space and machinery rooms, no further tempering of the air takes place after it leaves the fans, but in the case of all other areas where indirect radiation is employed, a thermostatically-controlled mixing damper is provided in each individual duct. These fans can thus deliver air through the many branches of their systems at various temperatures that are governed by the requirements of the areas served.

Unit Heaters

Comparatively large numbers of employes are grouped in several parts of the building, and to provide a plentiful supply of fresh air at these points, unit heaters have been installed which operate continuously along with a mechanical exhausting system.



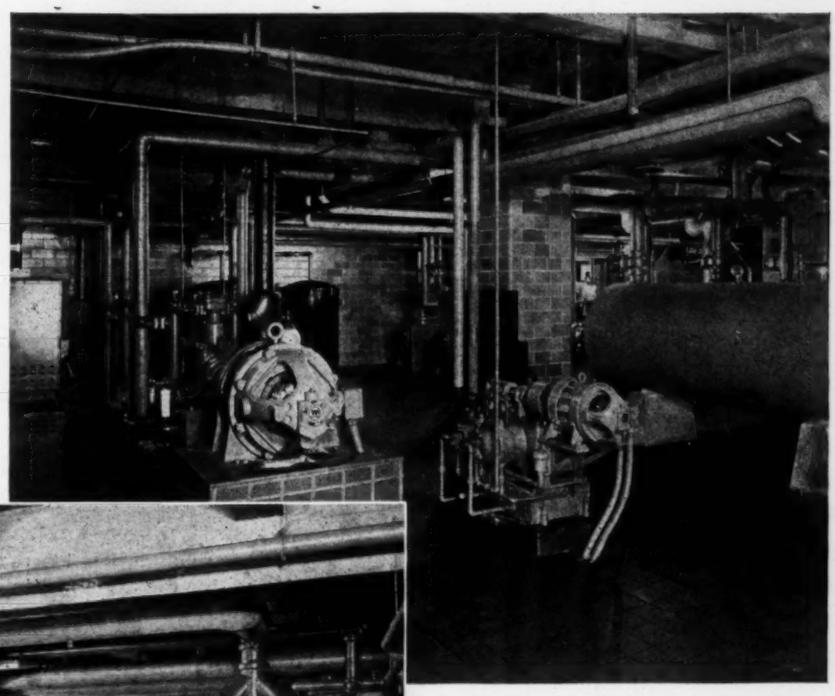
BELOW — MACHINERY ROOM— SUB-SUB-BASEMENT. SHOWING HOUSE PUMPS, WATER FILTERS AND FIRE PUMP

On the eighth and ninth floors, which are drafting rooms without partitions, cross ventilation is furnished by six heaters per floor, and single heaters similarly serve portions of the twelfth and thirteenth floors. Three more heaters are totally enclosed along one wall of the gymnasium, and on the fourth floor there are also three for the blue print room, tracing vault, and telephone operators' room. The capacity of the last mentioned is 600 c.f.m.—the others discharge 1500 c.f.m. All are capable of raising this air through 85 F, and are of the recirculating type.

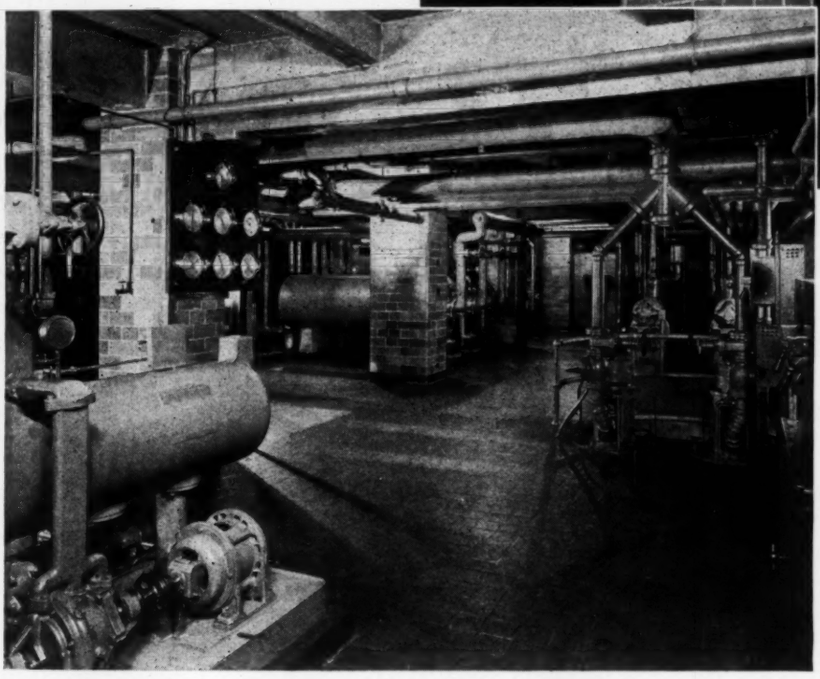
The gymnasium unit heaters have bottom fresh air inlets, as the windows are at a considerable height above the floor, but the inlet for all others is an integral part of the double hung windows. Bottom inlets for the latter would have meant placing an objectionable grille in the exterior spandrels of the building.

A brief description of the controlling mechanisms of this equipment will be of interest, as they embody several features which are new to this method of heating

and ventilating. Steam supply is automatically controlled by the same type of thermostatic valve used on the radiators, but the temperature at which the valve closes is slightly above that for which the radiator valves are set. This arrangement will prevent a condition in which frequent changes from cold air to heated air might otherwise exist if



BELOW — MACHINERY ROOM — SUB-SUB-BASEMENT. SHOWING VACUUM PUMP, WATER HEATERS, SEWAGE EJECTOR, AND AIR COMPRESSORS



ABOVE—MACHINERY ROOM—SUB-SUB-BASEMENT. SHOWING REFRIGERATOR COMPRESSOR, GENERAL SERVICE AIR COMPRESSORS, WATER HEATER AND AIR COMPRESSOR FOR STANDPIPE CUSHION TANK

steam were shut off the heating elements as soon as the fixed room temperature had been reached. It will thus be seen that the predetermined temperature of the room is maintained by the radiators during the heating season, while the unit heaters perform their function of supplying uniformly heated fresh air.

The danger of freezing up the heating element in very severe weather is apparent to all, and in order to avoid this the inlet damper is automatically closed by a power thermostat when steam is shut off the risers at night. When steam is turned on in the morning, this thermostat automatically opens the fresh air damper.

A second power thermostat completes the automatic control of unit heaters. At the beginning or end of the heating season there may be times when steam is on before the room temperature has dropped to a point where the steam supply to the unit heater cuts in. Should there be no necessity of heating the delivered air when this temperature finally is reached, the discharge damper will automatically by-pass the incoming air around the heating element. In normal operation, however, this damper will remain open, as the room temperature will be kept down by the radiators.

Water Heating

Hot water is supplied through three recirculating systems, one of which serves all floors above the twentieth, a second serves the basement cafeteria only, and the third supplies the remainder of the building.. All three heaters are located in the sub-sub-basement, those for the main building zones feeding up to headers that run along the ceiling of the twentieth floor corridor. From the lower zone header seven lines feed down through the building and return to their heater, while four risers for the upper zone extend to the top of the building and loop back to a return header on the twentieth floor. The cold water make-up lines connect with the main risers to the house tanks and maintain pressures at the heaters of approximately 240 lb. per sq. in. for the upper zone and 150 lb. per sq. in. for the lower zone.

Each of the three heaters is designed to heat 2,000 gallons of water per hour to 180 F with steam at one pound pressure and each is of 500 gallons storage capacity. As this heating capacity is well above the present demand, provision has thus been made, at a small additional first cost, to take care of any increased demand.

Ventilation

Due largely to the fact that nearly all offices have outside exposure on one or more sides, the natural ven-

tilation of the building is excellent, and, in addition to this, a considerable portion is also ventilated mechanically by exhaust and fresh air fans. The total cubical contents of the building amounts, as previously stated, to 6,462,000 cubic feet, and of this volume 2,300,000 cubic feet, or nearly 36 per cent, is ventilated by fans. A total of 200,000 cubic feet of air per minute is exhausted from the building, of which 118,000 is supplied by the fresh air fans and unit heaters.

The fresh air intake to the building is located on William Penn Way about 20 feet above street level. From this point the air is drawn by four fans at a velocity of approximately 900 feet per minute through an 11 foot by 9 foot insulated duct to the sub-sub-basement, where it first passes through oil filters which are automatically flushed with clean oil every twenty-four hours. The air next reaches the steam coils, after which it is discharged by the fans to the various floors.

To suit conditions that vary in different parts of the building, the aim in some areas is to build up a slight pressure, in others it is to create a vacuum, while in still others an even balance is held. The latter condition exists, for instance, in the sub-basement, sub-sub-basement, and shop space, where fresh air and exhaust each provide four and one-half changes of air per hour. The tendency in the drafting rooms, however, is to build up a pressure by means of excess fresh air, and thus avoid drafts, while still having good ventilation. Typical areas where a slight vacuum is aimed at are the toilets with twenty changes per hour and the cafeteria kitchen with thirty-five.

The tendency in the blue print room and elevator machine room is also towards a vacuum, where fifteen and seventy changes per hour occur, respectively, the purpose at these points being to remove heated air.

Another ventilated area which might be mentioned is the service entrance at the rear of the building. Three fan ventilators have been installed in the roof, and in order to avoid the danger of carbon monoxide poisoning from truck motors, if these should be left running with the entrance doors closed in the winter, a fan of 5,000 c.f.m. capacity exhausts the air at floor level, where the trucks back up to the loading platform.

With the exception of the fan just noted, all exhaust fans are located on the thirty-third floor, and discharge through louvered window openings at this level. Two shafts, one near each end of the building, enclose the duct risers and extend from the sub-sub-basement to the fan room, a total height of about 475 feet.

The fans are direct connected to constant speed motors running from 435 to 690 r.p.m., and discharge with a maximum velocity of less than 1,700 feet p.m., individual fan capacities ranging from 8,000 c.f.m. to 33,000. The motors range in size from 7½ to 20 hp. and are controlled by across-the-line starters.

Water Supply

Water is supplied from the city mains, and after being metered passes through sand and gravel pressure filters. The water pressure is about 100 lb. and the loss in the filters after a month's operation was five pounds.

Two recirculated drinking water systems are installed in the Koppers Building, the water being cooled to 45 F by the direct expansion of carbon dioxide. The direct connected compressor, of twelve tons capacity, is automatically controlled by a thermostat.

The structure is of fireproof construction and has, of course, automatic fire protection.

An underfloor drainage system was necessary under the sub-sub-basement.

Balancing Inside-Outside Conditions

By Malcolm Tomlinson

Article No. 2

IN the previous article of this title, published in the March, 1930, HEATING, PIPING AND AIR CONDITIONING, three charts were presented which enable the reader to obtain quickly approximate or actual effective temperatures for the range of air velocities usually experienced out-of-doors. Certain limitations, previously mentioned, should be kept in mind. Throughout these articles, the comfort data presented are averages which apply only to human beings normally clad, healthy and slightly active.

This article proposes to examine other experimental comfort data in order to determine, as nearly as possible, the adjustments which are necessary in an air conditioned room or building in order that people, entering or leaving, may not experience a sense of sharp discomfort. In other words, its purpose is to show how an air conditioned room inside a building may be "balanced" against adjoining unconditioned space. This is difficult, since the comfort data available represent test periods of from 15 minutes to four hours duration, whereas the time required to move from a conditioned to an unconditioned atmosphere is usually quite short.

One source of information pertinent to the subject under discussion is the data on the formation of sensible perspiration found in a paper on "Heat and Moisture Loss from the Human Body" published in the Journal of the A. S. H. V. E. (September issue) of HEATING, PIPING AND AIR CONDITIONING. These data are reproduced, except as to the wet bulb, in Fig. 1. Straight lines, to identify various stages in the formation of perspiration, connect the test data. The formation of perspiration apparently follows a fairly uniform rate for the body and for the forehead, although the two rates were different. This variation in effect appears to be due, in a large measure, to the clothing on the body and the exposure of the skin on forehead.

Evidently certain distinct zones of human discomfort, for high temperature conditions, are determined by perspiration formation. The zones in such formation may be designated as in Table 1.

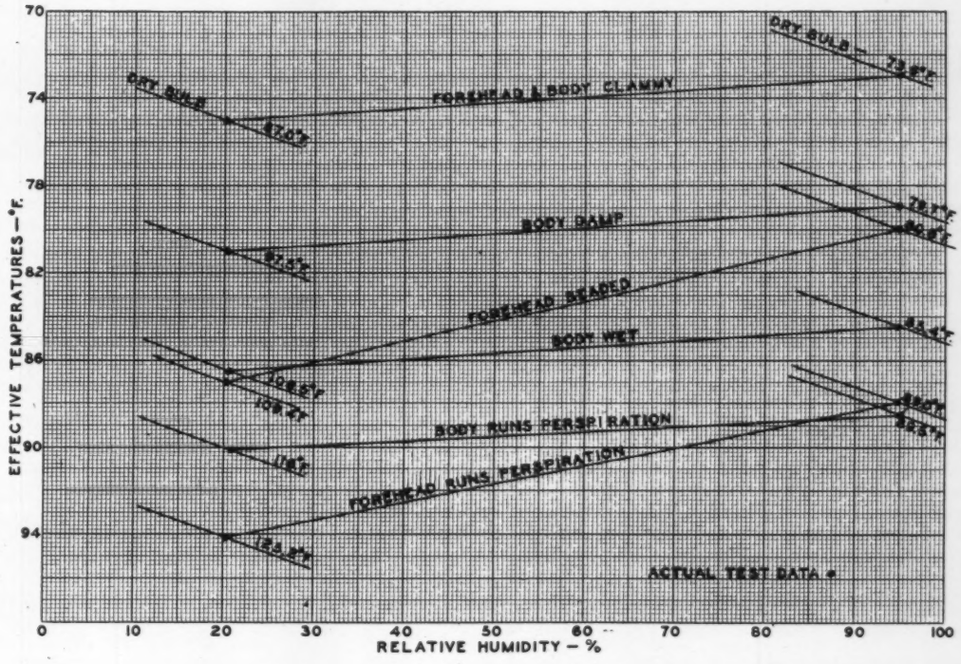


FIG. 1—FORMATION OF SENSIBLE PERSPIRATION

TABLE 1

BODY CONDITIONS	EFFECTIVE TEMPERATURE F	
	20% R. H.	95% R. H.
Clammy	75	73
Damp	81	79
Wet	86½	84½
Perspiration Runs	90	88½

Thus the "clammy" zone is 6 deg., the "damp" zone 5½ deg., and the "wet" zone 3¾ deg. wide in effective temperature, or a total of 15¼ deg.

When the heat lost and produced by the human body is plotted against effective temperature, as in Figs. 2 and 3, a zone is found in each case where the losses or the gains are constant under practically any condition of relative humidity. In this same zone the losses and gains apparently balance each other. If we plot the perspiration zones on these charts, it at once becomes apparent that the zone of balanced heat conditions is practically equivalent to double the width, in effective temperatures, of the total width of the three body perspiration zones so clearly defined in Fig. 1 and in Table 1.

The Harvard school of public health has studied the effect of climate and clothing on human comfort and has published information pertinent to the subject under discussion. In Table 2, the relation between the optimum indoor temperature and the diurnal and monthly outdoor temperatures is shown.

TABLE 2

PREVAILING OUTDOOR DRY BULB TEMP. F	PROBABLE OPTIMUM EFFECTIVE TEMP. F	MONTH	MONTHLY AVERAGE OUTDOOR DRY BULB TEMP. F
99-90	70.2	July	70.5
89-80	70.9	Aug.	67.4
79-70	70.4	Sept.	65.0
below 70	69.3	Sept.	65.0

The season variation in the comfort zone is shown in Table 3.

TABLE 3

COMFORT ZONE	EFFECTIVE TEMP. F			DRY BULB TEMP. F R. H. 50%		
	PROBABLE COLD LIMIT	PROBABLE WARM LIMIT	PROBABLE OPTIMUM	PROBABLE COLD LIMIT	PROBABLE WARM LIMIT	PROBABLE OPTIMUM
Summer	64	79	70.5	67.5	86.8	75.7
Winter	60	74	66.0	62.6	80.0	70.0

The influence of room occupancy on optimum temperature is given in Table 4 for 60 per cent relative humidity.

TABLE 4

NUMBER OF SUBJECTS IN ROOM	FLOOR AREA (SQ. FT. PER OCCUPANT		AIR SPACE (CU. FT.) PER OCCUPANT	MOST COMFORTABLE DRY BULB TEMP. F	PROBABLE OPTIMUM EFFECTIVE TEMP. F	OUTDOOR TEMP. F
	GROSS	NET				
8	44	20.5	380	75.0	70.8	77.0
25	14	7.5	120	73.4	69.4	81.9

In Tables 2 to 4, inclusive, the clothing worn was seasonal, so that the effect was normal in this respect. From this information, it is evident that crowded conditions as to human occupancy tend to lower the effective temperature; that the variation between the cold and warm limits of "perfect" comfort is 15 F for the summer, in terms of effective temperature; that the summer optimum effective temperature is 4½ F higher than for the winter and that the optimum effective temperature for the summer is practically a constant through an outdoor temperature variation of 29 F.

From our knowledge of the way in which the human body functions in adjusting itself to surrounding conditions, and of the Harvard comfort data just noted, it seems probable that the perspiration zones may vary in

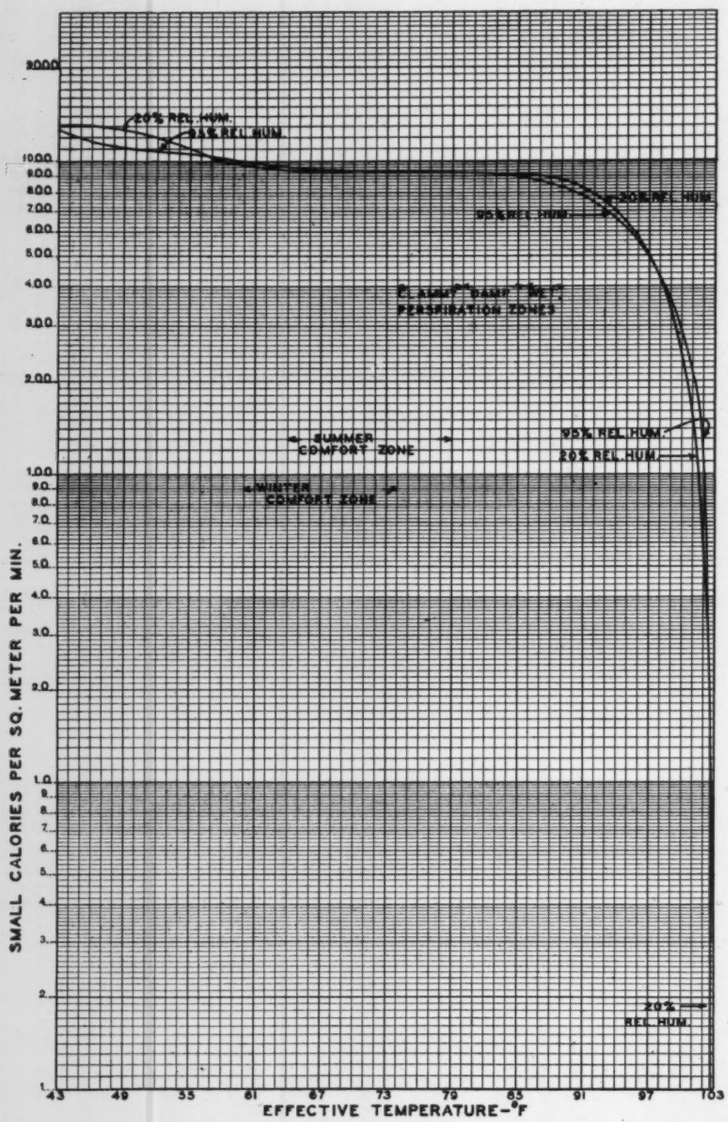


FIG. 2—RELATION BETWEEN HEAT LOSS AND EFFECTIVE TEMPERATURE

the different seasons of the year, although it is also probable that such variation is not great.

When the summer and winter comfort zones are noted on Figs. 2 and 3 and then compared with the perspiration zones also noted on these charts, it is evident that the summer comfort zone has a maximum effective temperature that is practically equivalent to the point where sensible perspiration begins.

From the information presented, it is logical to suggest that air conditioning equipment be operated in such a manner that inside or room conditions are balanced against outside weather by means of the effective temperatures rather than by dry bulb temperatures. Not only does the data presented show the definite relations between effective temperature and body characteristics, but it is evident to any student of human comfort that the time has arrived when dry bulb temperatures do not give sufficient information to be used alone in comfort considerations.

Since effective temperatures in the winter are very often as low as 45 F and, in the summer, frequently as high as 95 F, it would seem that a plenum chamber, or conditioned corridor, should be established in all plants where the balancing of indoor-outdoor conditions is needed. Here a mean effective temperature can be maintained. This plenum chamber would be located in every exit from a conditioned room to outside or unconditioned room atmospheres. In this way an outside atmosphere below or above normal, for the particular season prevailing, could be balanced against the conditioned inside room or building. Such a plenum chamber would be valuable where outside effective temperatures are below or above the 45 and 95 F limits. It might, in certain dehumidification installations, also prove useful in the maintenance of low relative humidities.

It is probable that the minimum and maximum effective temperatures for the proposed plenum chamber would be about 50 and 89 F. This would no doubt mean additional standby capacity for conditioning equipment, but the protection afforded persons who move in and out of the conditioned rooms, is, in many cases, ample justification for the investment.

Nothing unique or new will be found in these suggestions. Many of the best planned air conditioning installations are engineered along these lines—especially in theater work. At the same time, the comfort charts in Figs. 1 to 3, on pp. 220-222 in the March issue, will be found very useful.

A practical application will be given. Take, for example, a room with an effective temperature of 78 F and an outside atmosphere of 110 F dry bulb, 75 per cent relative humidity and 5 miles per hour air motion. From Figs. 1 and 2 (in the March issue) the outside effective temperature is estimated as 104 F. This is 25 deg. higher than the maximum for summer comfort. In this case, the plenum chamber should be operated at an effective temperature of about 90 F. For winter the minimum effective temperature is used.

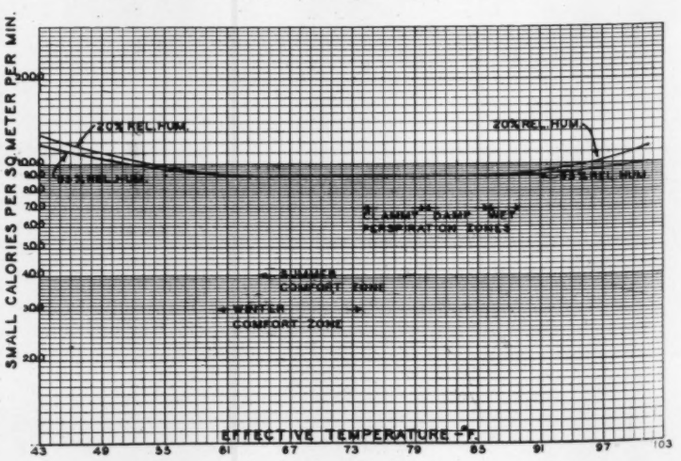


FIG. 3—RELATION BETWEEN HEAT PRODUCED AND EFFECTIVE TEMPERATURE

Drips and Drains

By G. W. Hauck

IN the design and installation of power plant piping, the drains or drips are often an after-thought, instead of being given proper initial attention. Consequently, they are often inadequate and, at best, partially effective. The purpose of this article is to show the faults of some popular methods of draining and further to show how easily efficient and practical drains can be installed.

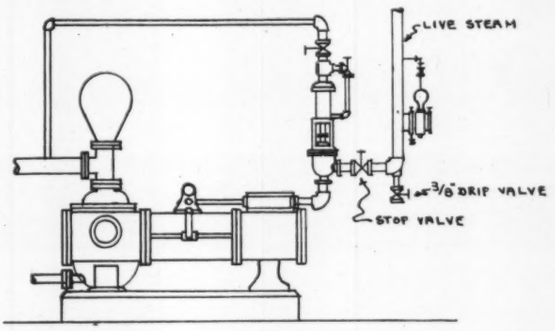


FIG. 1—SIMPLE METHOD OF DRAINING STEAM SUPPLY LINE TO PUMPS OR OTHER AUXILIARIES

Draining Steam Line to Auxiliaries

Fig. 1 illustrates an easy and simple method of draining the steam supply line to pumps or other auxiliaries. Where the stop valve is located in a vertical steam supply line, a small coupling can be welded in the piping immediately above the valve. In the larger units or auxiliaries it is advisable to install separators properly trapped and this figure shows a hand-operated drip for units using small piping. In starting up after a shut-down, this drip is especially useful in draining off the accumulated condensate.

Tapping Gate Valves

A standard practice of valve manufacturers now includes six bosses on all gate valves in the medium and extra-heavy iron body patterns and all steel valves. Two of these bosses are located on the bottom of gate valves and two on each side, as illustrated in Fig. 2. These bosses are of such diameter and height to permit tapping ½-in. pipe threads in 3-in. and 4-in. valves, ¾-in. threads in 5-in. to 8-in. valves and 1-in. threads in sizes 10 inches and larger. The location of these bosses permits valves to be tapped so that condensate can be drained with the valve in either the horizontal or vertical positions and drained on either side of the disc. In laying out piping, the designer should take advantage of these bosses and indicate tappings for drains. In certain installations requiring by-passes on gate valves, combination by-passes and drain assemblies can be furnished by the manufacturers. In boiler leads, where gate valves are located at the header, it is especially important that drips be installed to drain the condensate above the disc before opening such valve and cutting the boiler lead into the header. If condensate can collect above the disc of the automatic stop check valve on the boiler, this valve should likewise be provided with a drain.

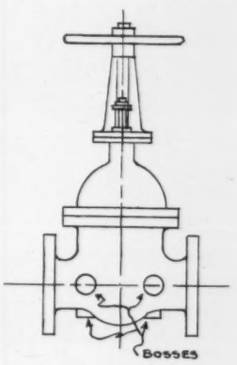


FIG. 2—BOSSES PERMITTING GATE VALVES TO BE TAPPED FOR DRAINING WHEN VALVE IS IN VERTICAL OR HORIZONTAL POSITION

Draining Headers

Fig. 3 illustrates a commonly-used method of draining headers. Tapped bosses are located on a number of tees and often the ends of headers are provided with a blind flange with eccentric tapping. While these tappings remove some condensate they usually are small and, with the higher velocities, condensate is carried past these small openings. A more desirable method of draining headers, the writer believes, is illustrated in Fig. 4. The use of drip pockets or pipe drip legs insures complete trapping of condensate even under the higher velocities. These drip pockets likewise provide a reservoir for slugs of water, should they occur. Cast one-piece drip pockets, as made by various manufacturers, can be used, or they can be made up with a piece of flanged pipe about eighteen inches long and with tapped blind flange, as shown at the end of header in Fig. 3. The cast drip pocket is often used, as it eliminates one pipe joint and is neat in appearance. These drip pockets should be the full size of header or steam line up to and including twelve-inch, or, at least, not less than one pipe size smaller. For lines larger than twelve inches, the twelve-inch drip pocket is large enough to take care of the condensate in steam lines from fourteen to twenty-four inches in size. These drip pockets aid also in keeping the piping clear of scale and dirt that can accumulate where the small drains shown in Fig. 3 are used. For those who prefer welded header and piping construction, the installation of welded drip pockets is suggested, as shown in Fig. 5. These welded pockets should follow the same practice as to size as outlined for the cast drip pockets mentioned before.

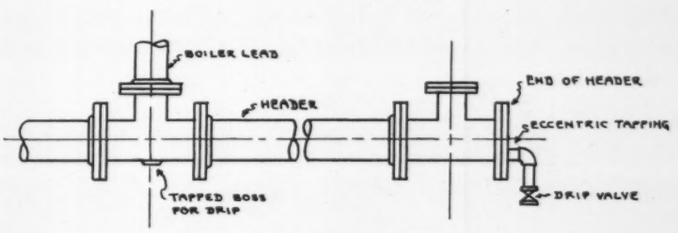


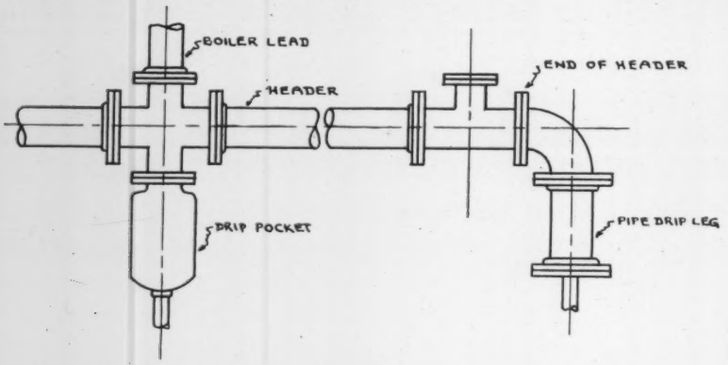
FIG. 3—COMMON METHOD OF DRAINING HEADERS

"This article contains excellent, practical ideas and the illustrations are good."—W. H. Wilson, maintenance engineer

Size of Drip Pockets

The drain tappings on drip pockets, either cast, or welded, should preferably be several sizes larger than the pipe connection on the traps. The piping

FIG. 4—A DESIRABLE METHOD OF DRAINING HEADERS



should be run full size to, and reducing at, the traps. It is obvious that the traps should be of the very best and efficient types to complete the function of draining, as a

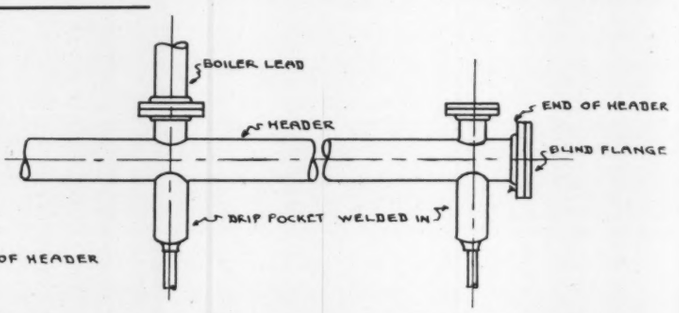


FIG. 5—INSTALLATION OF WELDED DRIP POCKETS

separator or drip pocket is dependent entirely for its effectiveness upon the ability of the trap to discharge the condensate as rapidly as accumulated.

In conclusion, note that the stress laid upon proper dripping or draining of steam lines is not based upon theory, but is proved in actual practice. Proper draining proves a safeguard, improves the operation of any steam plant, and the additional precaution of ample drainage is a profitable investment.

INTERVIEWS OF INTEREST

Health and Air Conditioning

Dr. W. J. McConnell discussed before the *American Institute of Mining and Metallurgical Engineers* in New York in February the manner of human response to changing atmospheric conditions. His discussion, which will be presented as Technical Publication No. 319 of the Institute, is of particular interest in these controversial days of ventilation interest because it presents for the first time many observations made during the course of Dr. McConnell's technical experimentation in Pittsburgh, still unpublished and unavailable except to the initiated few. Also, it gives a fully annotated, straight-line statement of the physiologist's path in evolving out of old ideas of noxious qualities of vitiated air to our present conception of the thermal balance requirements of the human body with the skin as the thermostat which sets up physiological adjustments against excessive heat or cold.

Healthy men are physiologically efficient under rather wide ranges of heat, humidity, and air velocity. The upper limits of physiological endurance has been determined at about 90 F, saturated, when the air is still. Low temperature reactions and endurance limits are far less definitely understood, and the relatively narrow range of greatest human efficiency has been given far less scrutiny than the subject demands. In a personal conversation with us, Dr. McConnell, of the Industrial Hygiene Service of the *Metropolitan Life Insurance Company,* declared:

"There is great need for an adequate ventilation standard. This we do not have. Elaborate and detailed studies have been made on the physiological effects of excessive and prolonged exposure to very high temperatures, but we have no parallel data on low temperature ranges.

"We know that effective temperatures fluctuate over a range of five to fifteen degrees. We do not know how long the cycle of such fluctuation should be.

"Neither do we know what is meant when the engineer is enjoined to maintain a given room temperature and relative humidity with 'suitable' air movement. No one knows what movement of air is suitable.

"Therefore, in future experiments to be set up, the effect of low as well as high temperatures needs to be studied. We need to know just what the 'efficient' conditions are for human action, and especially what are the effects of air motion of moderate rates, say from 50 to 150 lineal feet per minute, and what fluctuations in operation are beneficial or permissible.

"Moreover, it would appear to be appropriate to transfer the field of observation from school to industry. It is quite urgent to consider the air environment of office workers, for instance. Adult subjects are more stable than children. Contagious disease pictures are less likely to influence the general result of observations, and industrial situations are usually more capable of control.

"It has been useful to study the bad physiological effects upon limited groups of workers of air environ-

ments created for processes rather than for people. Knowledge thus gained can be used to combat unfavorable conditions, and work schedules are made possible of suitable adjustment. But these exigencies of environment are not the entire story, and it is even more important to find the happy solution of air environmental problems of the average man and to be able to prescribe that condition as a norm.

"Ventilation science has been afflicted with far too many single-theory men. The engineer and the physician alike now need to set about the serious business of proving up the physical basis of health claims and setting up new work on physiological fundamentals that relate to indoor acclimatization."

School Ventilation

H. W. Schmidt, superintendent of school buildings, state department of education, Wisconsin, has been making particular inquiry into actual physical conditions of school room atmospheres that obtain in buildings reputed to be well-ventilated. He discusses for us January tests just tabulated on a new and modern Wisconsin school building that is equipped with unit-room ventilation, installed with full conformance to Wisconsin state requirements on class room ventilation.

Two rooms on opposite sides of the building and fully occupied, were chosen for testing. The air exhaust through the ducts provided for the purpose was found to be about forty per cent of the primary air supply in one room and about thirty per cent in the other and there was definite leakage around the doors; the vent ducts became inoperative when the class-room door was opened. At the same time, infiltration increased, as shown by a Borozin machine. (This device makes use of a powder blown into the air to show the direction of the air currents.)

"From this, and other studies we have made, effective service from unit ventilation is noted. It is apparent that ventilators of the high velocity type introduce problems not met with in ordinary plenum or blast systems.

The Factor of Infiltration

"Also, it is apparent that in all systems the factor of infiltration plays a much more important role than is usually assigned to it and any device that tends to build up a static pressure and to cut down infiltration is eminently worth while," states Mr. Schmidt.

"We always have to make some allowance in our calculations for rooms on windward and leeward sides of a building, and increasing attention needs to be given to warm-weather ventilation. The desideratum is to provide abundant air at all seasons, with sufficient discharge to get rid of any odors and to remove bodily heat, at the same time maintaining a plenum pressure that will take care of undue infiltration, with control as nearly automatic as possible.

"The unit-room conditions tested meet these conditions with a high degree of success. Nevertheless, in unit as in any other system, one looks in vain for a self-steering ventilating job, or heating installation either; and everywhere, whatever the type of installation, an untrained personnel is a millstone that hampers the engineer and school administrator alike. It must be so until we reach a financial status that will permit increased salaries and a higher complement of trained men as school plant maintenance men.

"But conditions improve. We are not doing so badly as compared with conditions twenty years ago. The successful administrator has always kept at least four or five laps ahead of general opinion on technical subjects like school room ventilation, and the almost universal collaboration now between school men and engineering groups is a very hopeful sign. Even the most isolated rural school man has learned the fallacy of taking ventilation advice from the man who sells him hardware. And we are learning to base our ventilation opinions upon empirical data, carefully collected, rather than to follow ventilation theory and unproved 'systems.'"—*S. P. M.*

Some Fundamentals of Drying

DRYING is generally considered to cover two fields; one, the high temperature and the other, the low temperature. In the first field, the temperatures are above the boiling point while in the second field the reverse is true. Heat may be applied, directly or indirectly, by means of steam, hot water, hot air or electricity with fuels such as coal, waste gases, oil, or gases of natural or manufactured varieties.

Temperatures above the boiling point cause rapid evaporation. In the drying of materials containing little moisture, such as paints and enamels, high temperatures are used. Where such temperatures destroy the color or other physical and chemical characteristics, low temperatures are required. The latter condition is frequently met by a reduced pressure, as in vacuum drying, where the boiling point is correspondingly lowered.

Drying without Heat

Drying may also be accomplished without the use of heat. Through cooling mediums or refrigeration, moisture in the air may be condensed on chilled surfaces. Thus pipes through which cooling mixtures flow are frequently observed to drip water or to be covered with frost. With the aid of a fan to circulate the air, cold pipes will remove considerable moisture from moist air if the room is kept closed.

A second method of drying by cooling can be had by blowing moist air through canvas. Where the moist air encounters moving dry air on the opposite side of the canvas, evaporation is accelerated due to the cooling produced by the vaporization which takes place.

Use of Absorbent Materials

Another method of drying without the application of heat is in the use of porous and absorbent materials. Dry absorbent, or hygroscopic, materials take up moisture from the air. Charcoal, calcium chloride, sodium chloride (common table salt) and many other materials have this peculiar property. Cooling, and therefore drying, by the means of salts and other hygroscopic materials, is not efficient since, as no external heat is being supplied, heat for evaporation or vaporization is taken from the sensible heat of the liquid. As a result the temperature of the liquid drops and vaporization ceases. —*M. T.*

Along the Road of Controlled Indoor Atmosphere

Article No. 2

How can practical standards be established for control of indoor air? What temperature, humidity and air motion is required? Is ventilation needed; if so, where and of what character? Would air conditioning improve the efficiency of your plant?

OFFHAND, the heating of a building may appear to be a simple problem. When thoroughly analyzed, however, it is found to present a number of interesting sides which are well worth the consideration of the careful planner.

Good heating consists in the maintenance of an economically comfortable temperature throughout the occupied spaces, an economical process temperature where required, and enough heat elsewhere to prevent the freezing of water pipes and so forth. An important consideration in any of these cases is the maintenance of the required temperature where it is needed, with as little waste as possible in adjacent spaces where heat is not required. This is of great importance in the case of industrial or other buildings with high stories, where heat may be required at or near the floor, but on account of the natural tendency of heated air to rise there may be great overheating in the upper parts of the spaces with the consequent unnecessary loss of a great amount of heat through these upper parts of the walls and roof. Or again heat may be required in certain parts of a space only, but on account of natural radiation and convection, the whole space may be more or less heated, with the result that much heat is wasted. In such cases the problem is to control the heat so as to establish the required conditions where needed and waste as little elsewhere as possible.

When we speak of the temperature to which a space should be heated, we generally refer to the dry bulb temperature. This measure of temperature is of little value for determining the conditions of comfort for human beings or even for determining the proper atmospheric conditions for processes or products which require any refinement of air conditioning.

The Theory of Heating Standards

Recent research of the *American Society of Heating and Ventilating Engineers'* research laboratory, operating in conjunction with the *U. S. Bureau of Mines* and the *U. S. Bureau of Public Health* has shown that there are three important factors which go to make up the effective temperature, or the physiological temperature effect, which an atmosphere will exert upon human beings.

These are: the dry bulb temperature, the wet bulb temperature and the air motion.

The dry bulb temperature is that taken with an ordinary thermometer (usually reading in degrees fahr.) and represents the temperature of the air without reference to the effect of its moisture content.

The wet bulb temperature is taken with a thermometer, which has, in one type, its bulb encased in a silk mesh bag, the reading being taken after the bag has been moistened with clean water and after the instrument has been whirled through the air until the indication has become stable. The wet bulb temperature takes into account the conditions of the atmosphere as to its moisture content, due to the fact that the rate of evaporation of the moisture from the bag and the consequent cooling of the thermometer bulb, caused by this evaporation, depends upon the relative amount of moisture in the air.

By air motion is meant the velocity of the air in feet per minute. Its effect is that of a breeze in removing the enveloping air and bringing new air to take its place, thus producing a cooling effect by increasing evaporation.

The combined effect of these three factors is designated as the effective temperature of any particular atmospheric condition.

The Relative Importance of Temperature, Humidity and Motion

An idea of the relative importance of these factors may be gained by realizing that a dry bulb temperature of 70 degrees has an effective temperature of 60 degrees when the air is absolutely dry and without motion, an effective temperature of 50 degrees when dry and with an air motion of 500 ft. per min. and an effective temperature of 70 degrees when fully saturated and without air motion.

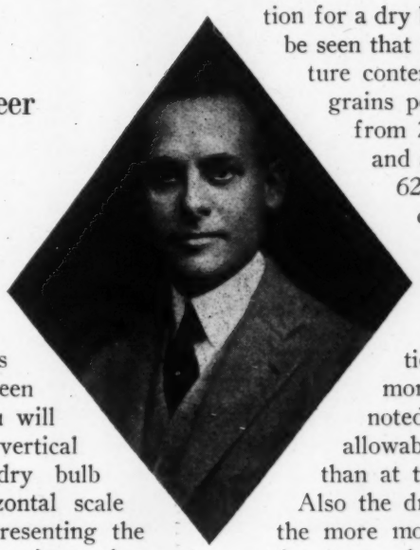
In the same way these factors have a similar bearing upon the effect that an atmosphere will have upon processes or products of manufacture. They determine the drying, cooling, heating, expanding, contracting, hydrating, dehydrating, cracking, warping, sweating and other effects and variations which any particular or varying atmospheric condition may exert upon the products.

The effective temperature and not the dry bulb temperature, nor the wet bulb temperature nor either or both, without the consideration of air motion, determines the heating requirements so far as the health, comfort and efficiency of workers or the quantity or quality of products are concerned.

The exact physiological effect of these factors has been established by the research laboratory of the *American Society of Heating and Ventilating Engineers,* as illustrated by the charts shown in Figs. 2, 3 and 4.

Fig. 2 shows the effective temperature chart for humans at rest in still air with different dry bulb and wet bulb temperatures corresponding to different percen-

By Perry West,
Consulting Engineer



tages of relative humidity, or moisture in the atmosphere. It might be said here that the relative humidity is calculated from the difference between the wet and dry bulb readings. You will note that this chart is made up of vertical isothermal lines representing the dry bulb temperature as shown on the horizontal scale at the bottom; horizontal lines, representing the grains of moisture per pound of dry air as shown on the vertical scale at the left; light oblique isothermal lines sloping downwards towards the right, representing the wet bulb temperatures as shown on the scale along the upper curved line; heavy oblique effective temperature lines, representing the effective temperatures as shown on the same scale as the wet bulb temperatures; and the oblique curved lines, representing the relative humidity in percentages of the quantity of moisture required to saturate air at the particular dry bulb temperature under consideration.

The shaded band between effective temperatures 62 and 69 covers the conditions of substantially equal human comfort and is designated as the comfort zone.

The effective temperature of 64 deg. is the optimum for the average human at rest in still air and is designated as the comfort line.

How to Use the Effective Temperature Chart

To use this chart start with the dry bulb temperature, say for example of 70 degrees, at *A* and follow the vertical line to its point of intersection with the wet bulb temperature of 57 degrees for this example, at *B*. The curved line passing through this point of intersection will represent the relative humidity, which in this case happens to be 45 per cent. Starting from *B* a line drawn parallel to the nearest effective temperature line to the scale of effective temperature will intersect this scale at *C*, the effective temperature reading, which happens to be 64 in this case.

Starting again from *B* a line drawn parallel to the nearest horizontal line will intersect the scale of moisture contents at *D*, the number of grains per cu. ft. of air, which happens to be 44 in this case.

Similarly by starting with any two known factors the others may be found.

Why Dampness Increases Discomfort

The example above represents the optimum effective temperature condition for a dry bulb temperature of 70 degrees. It will be seen that for this dry bulb temperature the moisture content may be varied between 24 and 100 grains per cu. ft. corresponding to a variation from 28 to 92 per cent of relative humidity and an effective temperature variation from 62 to 69 degrees, without getting outside of the comfort zone in either direction. It should be borne in mind, however, that this comfort zone covers the extreme limits for health and comfort and that the nearer the conditions are kept to the comfort line the more ideal they will be. It should be noted that around 70 degrees dry bulb the allowable variation in humidity is greater than at temperatures very far below or above. Also the dryer the air the hotter it must be and the more moist the air the cooler it must be, in order to produce the proper effective temperature. Fig. 3 represents the same kind of psychrometric chart with the comfort zone superimposed, for humans slightly active, in air moving at 300 ft. per min. and Fig. 4 the same, for air moving at 500 ft. per min.

It should be noted in connection with these two air-motion charts that a dry bulb temperature of 73 degrees with 300 ft. per min. air motion and 76½ degrees with 500 ft. per min. air motion are required to meet the comfort line requirements of 70 degrees dry bulb without air motion and with 45 per cent relative humidity in each case. Also that the widest allowable variation in humidity is around these temperatures. Also that, at temperatures above 55 degrees with 300 ft. air motion and above 60 degrees with 500 ft. air motion, the less the moisture in the air, the greater the cooling effect due to air motion, so that with saturated air at 80 degrees (dry bulb) the effective temperature is reduced 6 degrees and with dry air, 14 degrees with an air motion of 500 ft. per min.; whereas with 60-degree air the effective temperature is reduced 12 degrees with 500-ft. air motion and with 55 degrees air it is reduced 10 degrees by 300-ft. air motion, regardless of the amount of the moisture content.

Below 60 degrees with 500-ft. air motion and below 55 degrees with 300-ft. air motion the dryer the air the less the cooling effect.

At 50 degrees and 500-ft. motion the cooling for dry air is 13

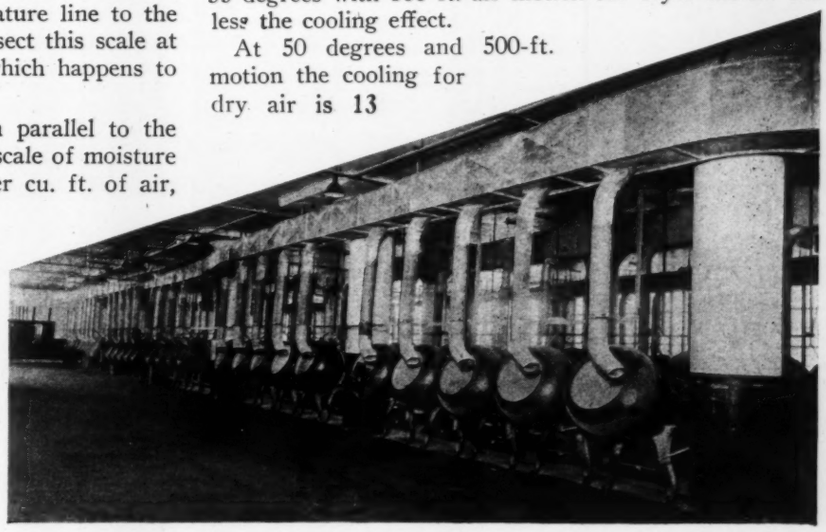


Fig. 1—Conditioned Air Applied to Coating Kettles in a Candy Factory

degrees and for saturated air this factor is 16 degrees.

At 45 degrees and 300-ft. motion the cooling for dry air is 10½ degrees and for saturated air is 18 degrees.

There are conditions, therefore, under which moisture in the air assists the cooling effect of air movements and other conditions under which it retards it. This is why one feels colder on a cold damp winter day and hotter on a hot damp summer day.

So far we have only discussed the conditioning pertaining to humans at rest or slightly active.

Comfort Conditions of Humans at Active Work

For those actively engaged in work there is considerable variation from the above conditions of comfort, depending upon the character and degree of the work. These conditions have not been very accurately determined as yet, but present indications are that for humans at hard work the effective temperature may be reduced at least 5 degrees so far as health and comfort are concerned. Some recent studies throughout the industries indicate that male workers show best results at a temperature of 72 degrees and a relative humidity of 40 per cent and female workers at 80 degrees and a corresponding lower relative humidity, with both conditions corresponding to an effective temperature of about 65 degrees.

It has been known for a long time that overheating is very enervating and greatly decreases the capacity of workers, but this condition cannot be diagnosed from the dry bulb temperature alone. If the humidity is decreased in proportion the temperature may frequently be increased with advantage, as shown by the above studies.

Studies of the *New York Commission on Ventilation* indicated that the capacity of workers is reduced about 28 per cent with a temperature of 78 degrees and a high relative humidity as compared with a temperature of 68 degrees and a low relative humidity.

This report is not definite, but the higher temperature and humidity would probably correspond to an effective temperature of 75 degrees and the lower conditions to about 63 degrees. It is fairly safe to say that an increase of each degree in effective temperature will decrease the capacity of workers about 2½ per cent.

The reason that the data on proper standards of heating are not more definite is that the physiological reactions upon humans is so different and indefinite that a great amount of very careful and laborious research is necessary to determine the exact conditions of comfort and vastly more to determine the insidious effects on health, efficiency and longevity.

When it comes to the effects upon processes and products the determinations are much easier and the available data much more exact.

For this reason it is sometimes found that factories for chewing gum, candy, tobacco, textiles and other products of manufacture are better heated and air conditioned than are the employes working in these factories.

Practical Standards for Heating for Humans Not Engaged in Manual Labor

The proper standard for heating where the employes are not engaged in manual labor is from 62 to 69 de-

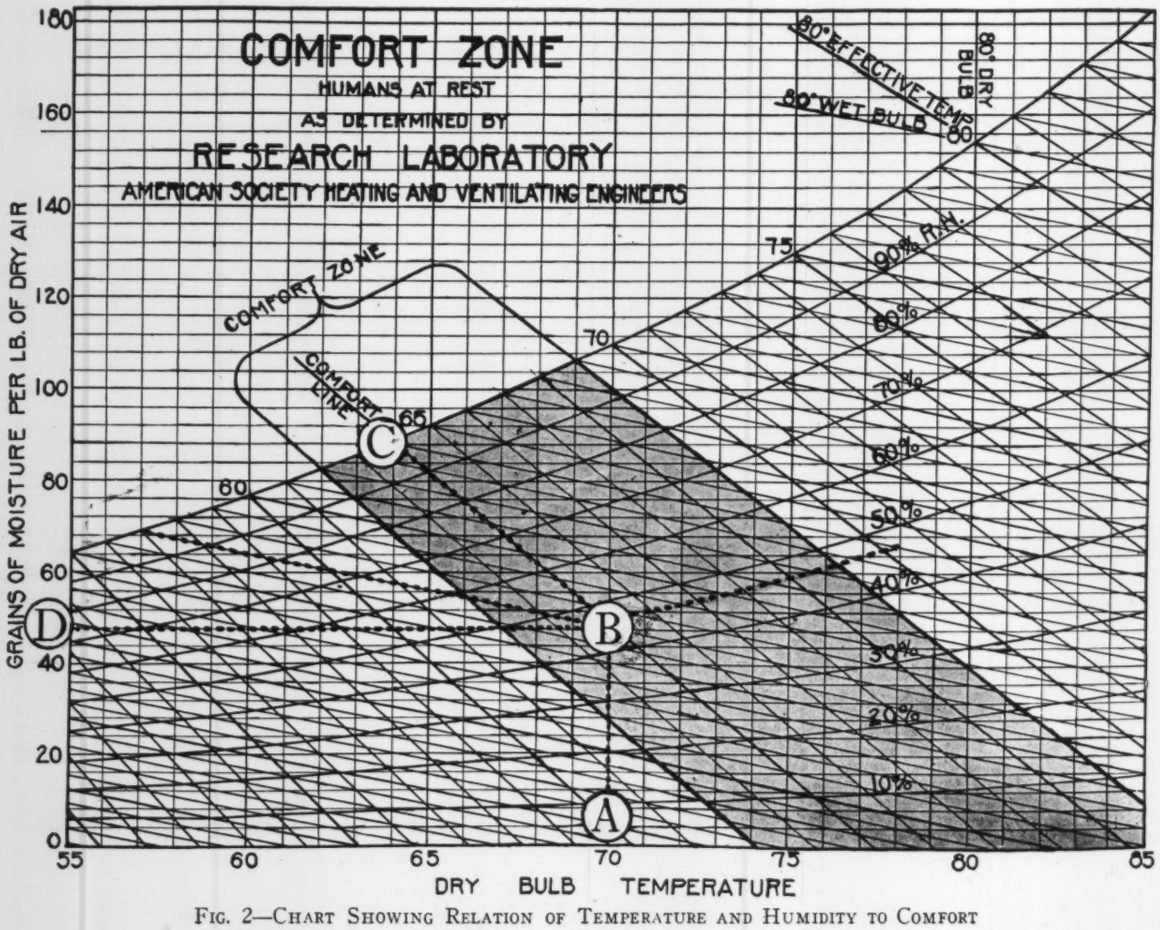


Fig. 2—Chart Showing Relation of Temperature and Humidity to Comfort

grees, with an optimum of 64 degrees effective temperature, but as pointed out above, the dry bulb temperature necessary to produce these results may be varied from 62 to 90 degrees, depending upon the relative humidity and air motion. Optimum conditions for all of these factors are perhaps around 70 degrees dry bulb, 45 per cent relative humidity and a slight air motion.

Where employes are performing light work, the dry bulb temperature may be reduced to 67½ degrees and where performing heavy work, to 65 degrees, with 60 degrees as the low limit. Departures from these standards will represent a loss in comfort and efficiency of about 2½ per cent per degree.

In producing these results it is necessary to take into account not only the heat to be supplied by the heating and ventilating equipment, but the body heat given off by the occupants themselves as well as the heat given off by the lighting equipment, motors and other apparatus, and properly balance these against the heat losses from the building.

In special cases where furnaces or other large heat emitting sources are present it is important to know the exact physiological effects which these will produce on the adjacent occupants as it is sometimes a problem of cooling by air motion or other means to prevent overheating at these points, and at the same time maintain sufficient heat elsewhere in the room. The kata-thermometer, which is an instrument which can be held at body temperature and at the same time record the rate of heating or cooling effect of surroundings, can be used to determine the conditions to be met in such cases.

What Do We Mean by Air Conditioning?

The term air conditioning may mean anything from the very best to the poorest and from the maximum to nothing. Air conditioning perfection for human health and comfort is a very definite thing which may be described as follows: that atmospheric condition in every part of indoor space occupied by human beings which is continually maintained with a proper amount of oxygen, free from dust, bacteria, objectionable odors, poisonous and other objectionable substances, with suitable air movements, and at the temperature and humidity quality shown within the zone of human comfort as defined by the joint research work of the *American Society of Heating and Ventilating Engineers,* the *United States Bureau of Mines* and the *United States Bureau of Public Health.*

Taking this as the highest aim of air conditioning and calling it 100 per cent we may then define the other extreme of 0 per cent of air conditioning as follows: that atmospheric condition in any part of indoor space occupied by human beings where one or more of the above enumerated factors is, or are, maintained so as to separately or collectively inflict death or permanent injury upon human beings.

Between these two extremes we have the wide range of conditions met with in practice.

Good air conditioning may be defined as follows: that percentage of perfection of the above factors which is warranted by the requirements of human health, comfort and efficiency on the one hand and expense and labor to produce these conditions (wherever they do not naturally exist) on the other.

Proper air conditioning for human health and comfort depends upon a number of factors which may be stated in the order of their importance as follows:

(1) Air Supply
(2) Air Temperature
(3) Air Cleanliness with reference to its freedom from dust and other suspended matters.
(4) Air Sanitation with reference to its freedom from bacteria.
(5) Relative Humidity
(6) Distribution
(7) Air Motion
(8) Freedom from Odors
(9) Freedom from other injurious substances.
(10) Freedom from monotony, with reference to noise and too much regularity of indoor conditions.

Theoretical Sandards of Air Conditioning

The results of history and experience have brought us to the present status of the art where it no longer is felt that the chemical composition of the air is the important factor but that proper ventilation and air conditioning depends more largely upon a number of other factors which may be stated in the order of their importance as follows:

1. Air supply.
2. Air temperature.
3. Air cleanliness in reference to its freedom from dust and other suspended matter.
4. Air sanitation with reference to its freedom from bacteria.
5. Relative humidity.
6. Distribution.
7. Air motion.
8. Freedom from odors.
9. Freedom from other injurious substances.
10. Freedom from monotony, with reference to noise and too much regularity of indoor conditions.

Air supply is put at the head of the list for the reason that while it no longer is considered to be the all important factor in ventilation, the amount of air to be supplied per person or the number of air changes to be furnished for any particular space will always be the starting point, for without air supply there can be no artificial ventilation.

The air supply is so vitally affected by the other factors mentioned that it cannot be determined independently and it will seem, that while this item is placed at the head of the list for the reason that it is the natural vehicle upon which the structure is carried, its importance beyond this point becomes subordinate to these other factors.

Air temperature is second for the reason that it has been proved by practically all of the accredited experimenters that overheating is more detrimental to the quality of ventilation than any other one thing.

Air cleanliness is third for the reason that it has to do

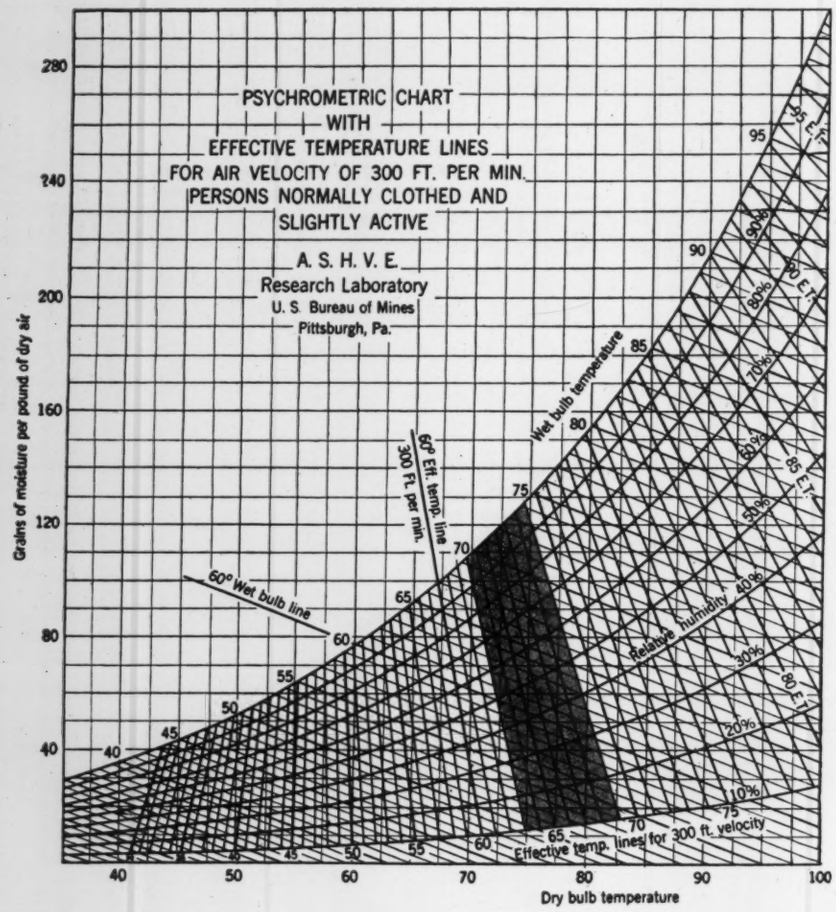


FIG. 3—PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE LINES FOR 300 FT. AIR VELOCITY. SHADED AREA INDICATES COMFORT ZONE

with human health both from the standpoint of freedom from dust and other suspended substances which irritate and clog the air passages, and from the standpoint of freedom from bacteria and other media of infection carried along with these substances which constitute the dirt in air.

Air sanitation is fourth, as it also affects human health and is correlated with the third item.

Relative humidity is fifth, not because it is of so much less importance than air supply and temperature but because it also bears such an intimate relationship with these two items that it receives a part of its due consideration in their determination. This factor will be further referred to in connection with air supply and air temperature in connection with which other factors are involved.

Distribution is sixth for a similar reason, for while it occupies a much more important place than its position might indicate, it is so intimately connected with the effective air supply that it receives a part of its consideration therewith.

Air motion is seventh in the same way, as it too receives a certain amount of its consideration in connection with effective temperature. Freedom from odors is eighth for the reason that while odors may become quite disagreeable and even nauseating they are seldom dangerous or permanently detrimental to health.

Freedom from other injurious substances is ninth, because these substances are so seldom found in ordinary ventilating practice and must be practically eliminated in any case.

Freedom from monotony is tenth because it has to do with the last refinements and the psychology of ventilation only.

The Function of the Synthetic Air Chart

In order to properly measure and compare the effects of these various factors the synthetic air chart as shown in Fig. 5 has been devised and adopted as the standard measure of the quality of ventilation.

The synthetic air chart takes air supply into account under the heading of CO_2. The scale for this factor is based upon the assumption that 300 parts of CO_2 in 10,000 parts of air, together with the other vitiation which would accompany this quantity of CO_2 when exhaled with the human breath, might produce results that would be permanently injurious to health. This is taken as the point where the quality of ventilation would drop to zero so far as this factor is concerned, and the scale between this and the point of perfection, where no CO_2 is present, is evenly divided so that for each part of CO_2 in 10,000 (above that ordinarily contained in the outside atmosphere and assumed at 4 parts in 10,000) a deduction of 1/3 per cent is made for this particular column, or department, and 0.3 per cent as the deduction in the final per cent of perfection column, from 100 per cent, which represents the point of perfection.

Air temperature together with air motion and relative humidity are represented under the column of effective temperature difference. These three factors are combined for the simple reason that the sensible or effective temperature depends not alone upon the dry bulb temperature but upon the relative humidity and air motion as well.

The scale for these factors is based on the assumption that an effective temperature of 104 deg. without air motion would soon cause permanent injury to health and this is taken as the point where the quality of ventilation for these factors would drop to zero.

Sixty-four degrees effective temperature was taken as the optimum point of human health and comfort without air motion and was assumed to represent 100 per cent quality. The scale is evenly divided between these two points so that each degree of difference between the observed effective temperature (after correction for air motion and condition of activity) and the ideal of 64 deg. represents a deduction of 2.5 per cent to be made from the 100 per cent for this column. This is taken to be equivalent to a deduction of 2.467 per cent per degree difference in the final percentage of per-

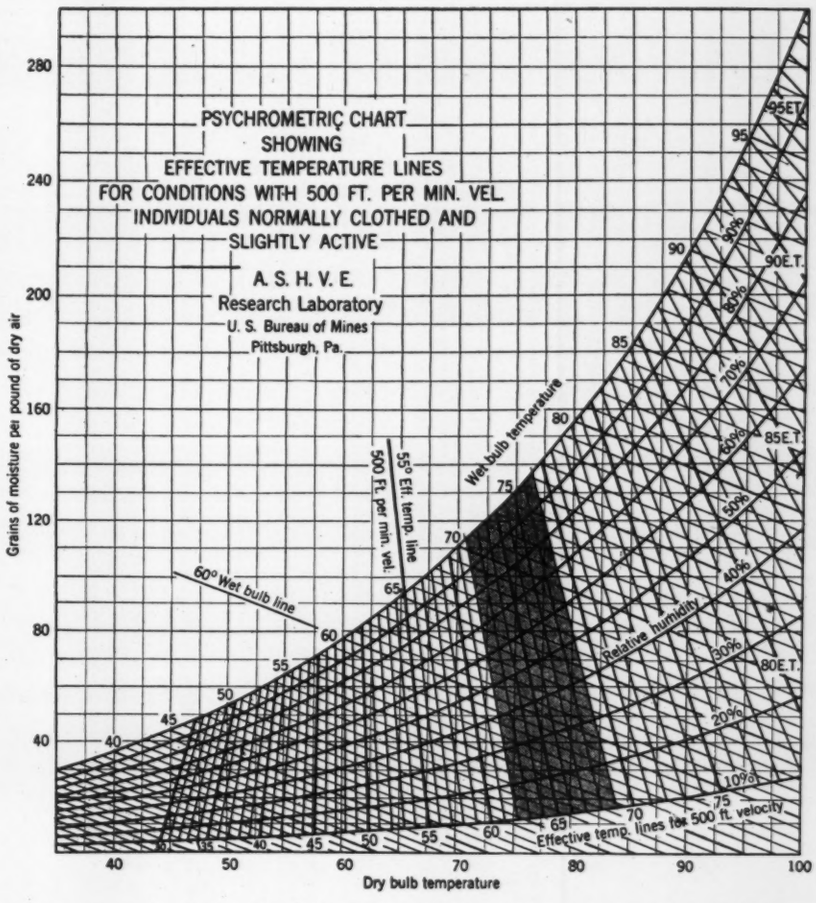


FIG. 4—PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE LINES FOR 500 FT. AIR VELOCITY. SHADED AREA INDICATES COMFORT ZONE

fection column (at the right of Fig. 5).

The dust column is arranged so that each 10,000 particles of dust per cu. ft. represents a penalty of 4 per cent for this particular column or department and 1 per cent in the final percentage of perfection column. The bacteria column is arranged so that each 5 colonies represent a penalty of 2 per cent for this particular column or department and 1 per cent in the final percentage of perfection column. The odors column is arranged so that each 10 per cent below the standard of perfection of freedom from odors represents a penalty of 1½ per cent in the final percentage of perfection column.

There is a separate column for distribution arranged with a scale imposing a penalty of 15 per cent for 50 per cent perfection of distribution and graduated so as to show a penalty of 1 per cent in the final percentage of perfection column for each 3 1/3 per cent of deficiency in the quality of distribution below perfection.

Finally there is a percentage of perfection column where the difference between the sum of all the penalizations and 100 per cent represents the composite percentage of perfection for the ventilation conditions represented.

Table 1, "Quality of air conditioning required for various kinds of buildings," shows what ventilation standards, according to this chart, should be maintained for the different requirements of spaces for which ventilation is ordinarily attempted.

Practical Standards of Ventilation

In the practical work of engineers who design ventilating systems and of architects and owners who have to pass upon them, the one item which is the basis of all calculations and layouts is the quantity of air to be handled. The functions of the air handled are, to supply the necessary oxygen for respiration, to keep the dilution of CO_2 and other objectionable substances down to the proper point, and to maintain the proper effective air temperature.

The air handled may consist entirely of air taken in from the outside or it may consist partly of new air and partly of recirculated air.

On the basis of oxygen supply and dilution only, Table 2 shows the cubic feet of air per minute required per person for the ventilation percentages shown, if all other factors are 100 per cent perfect.

In theaters, assembly rooms, auditoriums and other places of public amusement and assemblage there are usually several other factors to consider, such as the removal of excess heat, excess moisture, dust raised by the movement of the occupants and odors.

TABLE 1—QUALITY OF AIR CONDITIONING REQUIRED FOR VARIOUS KINDS OF BUILDINGS

	New Buildings %	Existing Buildings %
Schools		
Class Rooms	95	90
Manual Training Rooms	90	85
Domestic Science Rooms	90	85
Assembly Rooms	90	85
Toilet Rooms	85	80
Corridors	85	80
Churches	85	80
Hospitals		
Wards	95	90
Operating Rooms	98	93
Other Rooms	90	85
Theaters		
Seating Sections	90	85
Dressing Rooms, etc.	85	80
Dance, Lodge and Assembly Halls	88	83
Office Buildings		
Offices in office buildings or other buildings where persons are continuously employed	90	85
*Factory Buildings**		

*The percentage desirable for factory buildings will vary over a considerable range, depending upon the character of the work and of the process employed, modified to a considerable degree by the dust content of the air and the possibility of maintaining it free from objectionable dust and fumes.

An adult could exist indefinitely on the quantity of oxygen contained in 5 cu. ft. of air supplied per hour if the lungs could extract all of this oxygen. The practical percentage of extraction is placed at 25 per cent so that

TABLE 2—MINIMUM CUBIC FEET OF NEW AIR PER MINUTE REQUIRED PER PERSON FOR OXYGEN SUPPLY AND DISTRIBUTION ONLY

PERCENTAGE OF PERFECTION	CUBIC FEET OF AIR PER MINUTE REQUIRED PER PERSON AT REST	CUBIC FEET OF AIR PER MINUTE REQUIRED PER PERSON AT HARD WORK
98%	15.0	30
96%	7.5	15.0
94%	5.0	10.0
92%	3.75	7.5
90%	3.0	6.0

20 cu. ft. of air per hour will sustain life. An adult at rest expires about 0.6 cu. ft. of CO_2 per hour so that with 20 cu. ft. supplied per hour the CO_2 content would be 300 parts in 10,000. This equals the 300 parts fixed by the synthetic air chart for zero ventilation.

In the normally crowded city place of assemblage, the heat given off by the occupants together with that given off by the lighting and power equipment is usually more than the normal heat loss through the structure to the outside air, even in winter under cold climatic conditions. This means that in order to preserve an equilibrium of effective temperature, the entering air must be cooler than the leaving air, so that the problem is usually one of cooling and ventilating rather than of heating and ventilating.

A typical case for winter might show about 300 B.t.u. of body heat plus 100 B.t.u. from light, etc., being given up to the building against 200 B.t.u. heat loss from the building per person per hour. This would mean that 200 B.t.u. per person must be carried away by the air.

If the flow of air is upward, or from the side, so as to bring the incoming air into direct contact with the occupants, the temperature of the incoming air may not be more than 5 deg. below the temperature of the air leaving the occupant (for ceilings 10 ft. or less in height), otherwise the ventilation will be drafty and uncomfortable. This difference may be increased 1 degree for each 2 ft. of added ceiling height provided the rising air does not come into direct contact with another tier of occupants. For ceilings 10 feet and lower, the quantity of air per person to dissipate this excess heat is $200/60 \times 0.02 \times 5 = 33$ cu. ft. per person per minute. This may be reduced somewhat on the assumption that the component of heat from lights is usually introduced near the ceiling and may be allowed to heat the outgoing air to a greater difference.

Cooling for Warm Summer Weather

For 85-deg. outside air and 70 per cent relative humidity in the summer the effective temperature percentage of perfection would drop to 60 per cent without any change in the air from outside conditions. Assuming that the distribution was 85 per cent perfect, the total deduction would be 40 per cent plus 4.5 per cent, leaving a plus percentage of 55.5.

Assuming that the air supply is 30 cu. ft. per person and that the body heat and heat from equipment will raise this 5 deg.; also that the vapor added per person is 10 grains per minute or 0.33 grains per cu. ft. of air handled, the effective temperature difference will be raised about 2½ deg. so that the percentage of ventilation would drop about 6 per cent more, leaving 49.5 per cent ventilation. It will be seen, therefore, that we may not hope to get better than 40 to 50 per cent ventilation in warm summer weather without some method of air cooling.

Air motion will assist but unless increased beyond the usual 10 to 20 ft. per min. ordinarily obtained from the movement of the air through the room it will not improve the percentage of ventilation more than 2 to 3 per cent.

By the use of refrigerating and dehumidifying apparatus this effective temperature department can be maintained at any desired percentage of perfection. The use of a good air washer should reduce the temperature about 70 per cent of the difference between the wet and dry bulb temperature. This for the above case would reduce the effective temperature difference about 2 deg., corresponding to an increase of 5 per cent in the final percentage of the ventilation.

It will be understood that the above is rather an extreme case of temperature and humidity and that the final percentage will be improved by the air washer in a greater proportion if the relative humidity of the outside air is lower.

For a condition of 80-deg. dry bulb and 50 per cent relative humidity, the percentage for the entering air would be 79 per cent, the percentage leaving the occupants would be 67.5 per cent and the air washer would improve this to 76 per cent.

Assuming that 90 per cent ventilation is desired for places of assemblage, that distribution will be 95 per cent, dust 99 per cent, bacteria 99 per cent and odors 85 per cent, there will be a deduction of 1 per cent for dust plus 1 per cent for bacteria plus 2.3 per cent for odors, making a total of 4.3 per cent and leaving a deduction of 5.7 per cent for effective temperature plus distribution and CO_2.

Assuming that the effective temperature can be controlled in winter to within 1 deg. above or below the comfort line, there would be a deduction of 2.7 per cent for this, leaving a deduction of 3 per cent for distribution and CO_2. With 95 per cent distribution this would leave 1½ per cent reduction for CO_2, corresponding to an air supply of 20 cu. ft. per min. per person.

For a summer condition of 80-deg. dry bulb and 65 per cent relative humidity without air cooling, the effective temperature percentage would drop to 74 per

The research laboratory of the American Society of Heating and Ventilating Engineers has demonstrated that the significant factor in providing a comfortable condition for human beings is the effective temperature. Not only is the human body affected by this factor; exactly the same laws govern the final effect of thermal phenomena upon any body capable of exchanging both water and heat with the air surrounding it.—Edgar S. Miller, in the February HEATING, PIPING AND AIR CONDITIONING.

cent for the entering air and 62 per cent for the air leaving the occupants. This would be raised to 66 per cent with the air washer.

For 30 cu. ft. per person per minute these percentages would change to 74 per cent for the entering air, 66 per cent for the air leaving the occupants and 76 per cent with air washers.

For 40 cu. ft. per person per min. these values would change to 74 per cent, 67 per cent and 80 per cent and for 50 cu. ft. per person to 74 per cent, 68 per cent and 81 per cent.

It will be seen from this that 10 cu. ft. of air taken in from the outside, per person per minute, is sufficient for winter conditions but inadequate for summer climates, where heat and humidity are the determining factors, unless refrigeration is used. It should also be noted that increasing the air supply from 20 to 50 cu. ft. per person per min. gives little improvement unless some form of artificial cooling is used.

It would seem then that about 75 per cent ventilation can be obtained under reasonably severe summer conditions with an air supply of 30 cu. ft. per person, using an air washer, and that beyond this point there is little to be gained by increasing the supply. It should be noted that the above is based upon upward ventilation and that the cooling effect of from 5 to 10 deg. with air washers and of perhaps twice this amount with refrigeration may produce uncomfortable drafts on the occupants at times. For this reason and for the further reasons that it is, as a general rule, more easily controlled, the downward system of ventilation is often considered for large and intensely used places of assemblage. On account of transporting all of the heat from lights downward and of forcing the body heated air back over the occupants it is usually necessary to do much more cooling of the air than can be done with the air washer without refrigeration.

On the other hand the air is brought in high enough to permit of its being diffused and brought to the proper condition before coming in contact with the occupants. It can be seen, therefore, that the air supply per person per minute for assemblies could be 20 cu. ft. in winter, 30 cu. ft. in summer with air washers and anywhere between these two figures for the entire year with refrigeration. Also that nothing better than about 75 per cent ventilation can be secured in hot sultry summer weather without artificial cooling but that with cooling, especially if the air supply is taken from overhead and exhausted from below, most any percentage of perfection can be maintained.

Recirculation Considered

A good arrangement is to provide apparatus for handling 30 cu. ft. of air per person per minute with provisions for recirculating any amount up to as much as 2/3 of this. The percentage of air recirculated may be varied to suit the seasonable change so as to conserve heat in winter and refrigeration in summer.

Where effective temperature is controlled, according to the usual method, from the dry bulb temperature in the room, there may be a wide variation in this effective temperature due to the varying amounts of moisture in the air, unless humidifying apparatus with accurate humidity control is employed.

Between the condition of absolutely dry air at 70 degrees and absolutely saturated air at 70 deg. there is a difference of 10 deg. in effective temperature which means an average difference of 25 per cent in the quality of ventilation. This may be taken to mean about 10 per cent on each side of the neutral point for ordinary

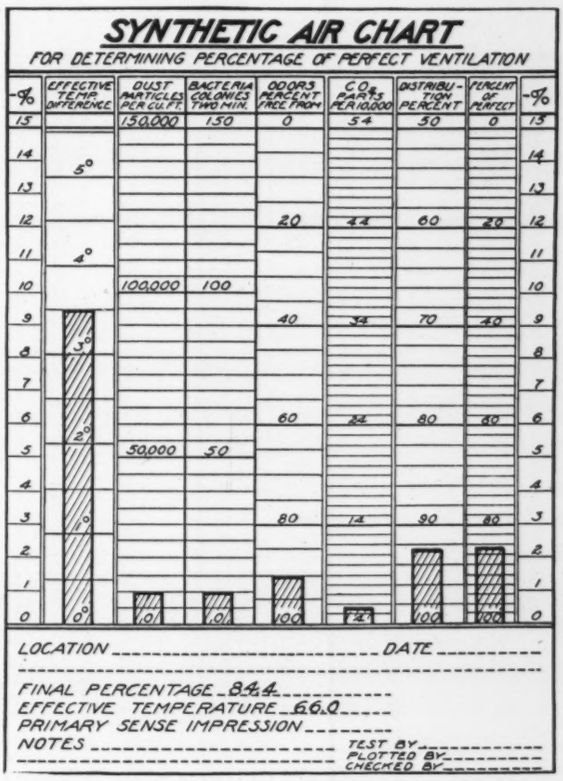

SYNTHETIC AIR CHART
FOR DETERMINING PERCENTAGE OF PERFECT VENTILATION
% EFFECTIVE TEMP. DIFFERENCE DUST PARTICLES PER CU. FT. BACTERIA COLONIES TWO MIN. ODORS PERCENT FREE FROM CO2 PARTS PER 10,000 DISTRIBUTION PERCENT PERCENT OF PERFECT %
15 150,000 150 0 54 50 0 15
14 5°
13
12 20 44 60 20 12
11 4°
10 100,000 100
9 40 34 70 40 9
8 3°
7
6 60 24 80 80 6
5 2° 50,000 50
4
3 80 14 90 80 3
2
1 1°
0 0° 10% 10% 100 FAIL 100 100 0
LOCATION DATE
FINAL PERCENTAGE 84.4
EFFECTIVE TEMPERATURE 66.0
PRIMARY SENSE IMPRESSION
NOTES
TEST BY
PLOTTED BY
CHECKED BY

FIG. 5—CHART FOR RECORDING RESULTS OF VENTILATION TESTS

ventilating conditions so that the air washer and humidity should improve the ordinary ventilating plant another 10 per cent on this count.

It should be noted in connection with our present means of measuring and comparing qualities of ventilation that we do not take into account any of the functions of the relative humidity of the air except that bearing upon effective temperature. This means that air of any temperature and relative humidity, within proper physical range, i. e., below 64 deg. wet bulb may be made to meet the comfort line by either heating or cooling without addition or deduction of moisture. Absolutely dry air may be heated or cooled to 78 deg. and be 100 per cent perfect as far as effective temperature is concerned and still be far from perfect as far as effects on the membranes of nose, throat and lungs are concerned. Such dry air is also very conducive to the increase of dustiness in the atmosphere of a room from the standpoints of dryness and electro-static agitation. The air washer and humidifier correct these difficulties and there should be some definition of limits for the relative humidity in our measure of ventilation.

Dust as a Factor in Ventilation

It is not unusual to find from 1 to 2 million particles of dust per cu. ft. in the outside air surrounding city buildings and unless this is eliminated it will give dust counts

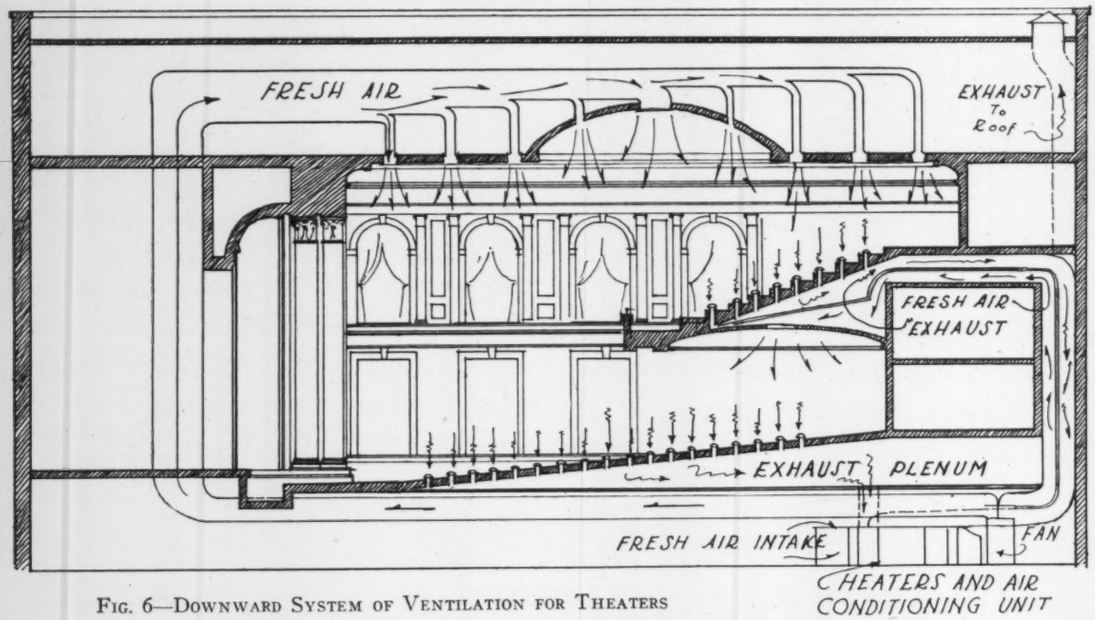


FIG. 6—DOWNWARD SYSTEM OF VENTILATION FOR THEATERS

in rooms equivalent to a deduction of from 5 to 20 per cent in the perfection of ventilation.

A good air washer should eliminate 80 to 90 per cent of the dust entering the intake and perhaps reduce the dust penalty in the rooms to less than one half of the above figure. It will be seen, therefore, that air washing and humidification may improve the quality of ventilation about 10 per cent in the effective temperature department, plus another 10 per cent in the dust department, plus other improvements in the quality of ventilation by maintaining proper humidity and removing other injurious substances and odors. Good filters will of course serve the same purpose for cleaning the air of suspended matter, and may improve the ventilation about 10 per cent.

Kitchen and lunch rooms should be ventilated with about a 5-minute air change for lunch rooms and a 1- to 3-minute air change for kitchens with at least a part of the exhaust taken from the lunch rooms through the kitchen so as to keep all of the air traveling towards the kitchens, thus preventing the kitchen odors from diffusing into the lunch rooms.

Ventilation of Hospitals

In the ventilation of hospitals the wards may be treated much the same as the class room of a school with the exception that it is inadvisable to use recirculation on account of the danger of contagion.

Dining rooms, kitchens and diet kitchens should have double mechanical ventilation, giving from 3- to 5-min. air changes for dining rooms, from 1- to 3-min. changes for kitchen, with the exhaust about 20 per cent in excess of the air supply with at least a part of the dining room exhaust passing out through the kitchen so as to keep all of the air travel toward the kitchen and prevent diffusion of odors throughout the buildings. Dining rooms, kitchen and toilet exhaust should be discharged above the roof of the building.

Exhaust from kitchen range hoods should be discharged by a separate exhaust fan through a fireproof metal duct extending above the roof of the building provided with automatic fire damper and steam jet fire extinguisher for use in case of emergency, as the accumulation of the grease vapors in this flue frequently causes a fire.

Private rooms if ventilated should be provided with double mechanical supply and exhaust ventilation as either a supply without exhaust or exhaust without supply will tend to force or draw the contaminated air from one room to another. Hotels, kitchens, restaurants and

TABLE 3—CUBIC FEET OF NEW AIR TO BE SUPPLIED PER PERSON PER MINUTE

	Without Humidifcation or Recirculation	With Humidification, Dehumidification but Without Recirculation	With Humidification, Dehumidification and Recirculation	Number of Air Changes Per Hour
Schools				
Class rooms	30	20	5 to 10	.....
Assembly rooms	15 to 20	10 to 15	5 to 10	.....
Gymnasiums	30	25	15 to 20	.....
Toilets	.....	.....	.....	10 to 20
Locker rooms	.....	.....	.....	5 to 10
Kitchens	.....	.....	.....	20 to 60
Lunch rooms	.....	.....	.....	10 to 20
Theaters				
Seating space	30 to 50	20 to 30	10 to 15	.....
Hospitals				
Wards	40 to 60	30 to 40	.....	.....
Kitchens	.....	.....	.....	20 to 60
Dining rooms	.....	.....	.....	10 to 20
Toilets	.....	.....	.....	10 to 20
Hotels				
Dining rooms	.....	.....	.....	10 to 20
Kitchens	.....	.....	.....	20 to 60
Ball rooms	20 to 30	15 to 20	10 to 15	5 to 10
Work space	.....	.....	.....	5 to 10
Assembly rooms	20 to 30	15 to 20	10 to 15	.....
Offices	20 to 30	15 to 20	5 to 10	.....
Work rooms	20 to 50	10 to 25	5 to 15	.....

dining rooms should be ventilated the same as for hospitals.

It would be interesting to analyze the ventilation for other classes of buildings, if time and space were available, but in lieu of this, Table 3 is submitted in an endeavor to outline the kinds of spaces usually requiring ventilation in various classes of buildings, together with the quantity of air ordinarily used for securing good ventilation results.

Standards for Air Conditioning

The standards for air conditioning are based upon the same factors of wet and dry bulb temperatures, air motion and air cleanliness, the most suitable value for all of these factors being well established for any particular process or product of manufacture.

Unlike the problem of ventilation the conditioning of air for manufacturing processes does not depend upon the indefinite physiological reaction of human beings, but upon the definite operation of physical laws in connection with the characteristics of materials.

We shall not attempt to give a table covering the extensive field of these requirements, and the best conditions suited thereto, but any competent specialist in this line can very quickly determine what conditions are best suited to any particular requirement.

Once the required conditions are ascertained it will be found that equipment is available in the market for producing just the kind of controlled indoor atmosphere best suited to any process of manufacture or other industrial requirement. Numerous articles in this publication have shown what these conditions are.

How to Weld Wrought Iron Pipe

Due to its distinctive method of production, genuine wrought iron has properties that are quite different in many respects from low carbon or mild steel, with which it is sometimes confused. Until recently wrought iron was made almost exclusively by the puddling process. In this process a charge of pig iron is melted in a special reverberatory furnace and iron oxide is added to remove the sulphur, phosphorus and carbon in the pig iron. As the carbon is eliminated, the metal becomes pasty since the melting point of the pure iron formed is considerably higher than that of the pig iron. The purified iron is removed from the furnace in balls weighing about 400 lb. and containing considerable of the slag formed during the purifying operation. The excess slag remains in the metal, however, and during the finishing operations this slag is distributed throughout the metal. As a result, genuine wrought iron has a fibrous structure which distinguishes it from steel.

The puddling process requires strenuous hand labor during the refining operation and is consequently not well adapted to quantity production. Recently, however, a process has been developed that eliminates hand puddling and permits increased output. Molten pig iron, completely refined in a modified Bessemer, open-hearth or electric furnace, is mixed with molten slag of suitable composition. As the slag has a lower melting point than the iron, when the two are mixed in correct proportions, the iron is solidified into a spongy mass which is permeated throughout with the liquid slag.

Welders should be familiar with the properties of wrought iron as it may be encountered in a variety of forms, the most important from the point of view of welding being wrought iron pipe. Important tonnages of this pipe are manufactured every year.

The general technique for welding wrought iron pipe is quite similar to that for welding steel pipe. Such slight modifications as are necessary are due almost entirely to the fact that wrought iron contains a certain amount of slag. No change is necessary in preparing the pipe ends for welding, in lining up or in the method of making rolling or position welds.

A puddle of molten metal should always be maintained at the point where rod is being added, the end of the rod being kept immersed in this molten puddle until enough metal has been deposited. Then the edges of the molten puddle and the surrounding colder metal should be fused together to obtain a solid union. The weld progresses by repetition of this procedure, care being taken to add no metal from the welding rod except through the vehicle of the molten puddle. In order to obtain sound weld metal free from oxide, which produces low strength and porosity, the molten puddle should be disturbed as little as possible. Excessive rubbing or agitating of the molten pool should be avoided as this causes undue exposure of the molten metal to the atmosphere, resulting in the formation of oxide which is frequently trapped in the weld.

With reasonable care during the welding operation, impurities easily can be floated to the surface. Undue agitation of the puddle may also result in molten metal being spread out over un-fused surfaces causing laps or other defects in the weld.

In welding wrought iron for the first time, the slag may tend to mislead the welder as to the fusion of the metal. The slag is disturbed throughout the metal and is more easily fusible than the iron. Accordingly when the blowpipe flame is applied to the surface of wrought iron the slag melts first, giving the appearance of fusion before the metal has actually melted. Once recognized, this condition is easily avoided by simply making certain that the sides of the vee are actually fused before adding weld metal. It should be remembered that wrought iron can stand higher heat without damage than can other ferrous materials and in general should be worked hotter for best results.

Molten wrought iron is more viscous than molten steel and consequently does not flow quite as readily. This is an advantage in welding pipe as it facilitates thorough and correct fusion at the bottom of the vee.

Another advantageous welding characteristic of wrought iron is the freedom from sparking which results from its very low carbon content.

Trap Troubles*

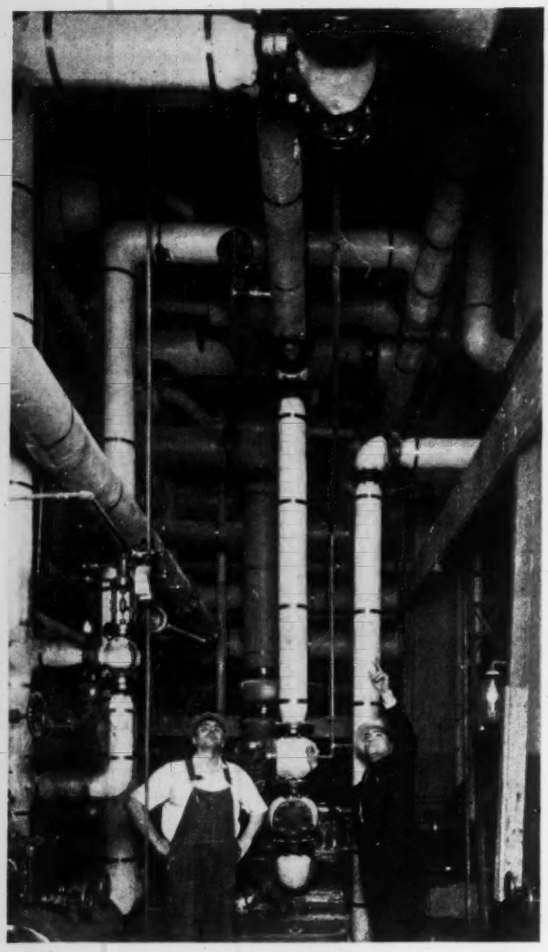


FIG. 21—PUMP ROOM PIPING ON BOILER ROOM FLOOR IN A LARGE HOTEL

RECENTLY a modern laundry establishment purchased two steam traps from a well known manufacturer. A few days after the traps were delivered, the laundry owner phoned the trap factory and said, "One of those traps you sent us is working fine but the other one doesn't seem to be any good. Will you send over a service man?" When the service man called he found that both traps had been installed upside down. One trap was blowing steam, which was considered satisfactory as the machine it drained was very hot. The valve orifice in the other trap had become filled with dirt and the machine was cold. As usual the traps were blamed, although they were both mechanically perfect.

Consider another case: A trap installed on the siphon-drained cylinder of a milk dryer would not keep the cylinder hot. When the factory service man inspected the job, he found that the siphon pipe had broken off and, naturally, the cylinder had to be half full of water before any water could get to the trap.

The installation shown in Fig. 22 was found when a factory service man called to check a trap that would blow steam the instant the steam valve was opened. For some unknown reason, line *A* had been put in but when the engineer installed the trap he entirely overlooked the bypass. As soon as the pipe was removed the trap operated perfectly.

* This is the last of a series of five articles by T. H. Rea on the fundamentals of trap installation and operation.

An eastern manufacturer reported that he could not get good service from the traps on some bucket-drained revolving cylinders. The cylinders were about half full of water all the time and as there were no siphon pipes to break off, the traps were blamed. Fig. 23 shows how the buckets fastened to the cylinder head pick up water from the bottom and after rotating 90 deg. in the direction indicated by the arrow, the water runs down into the trough connecting to the discharge pipe.

Further examination showed that these cylinders had been revolving in the wrong direction and hence no water could get out until the level was above the trough leading to the discharge pipe.

Trouble cases such as these are not as uncommon as might be imagined. They are merely cited to show that just because a heating unit is not working right, it is not positive proof that the fault is with the trap. On the contrary, there are more different kinds of trouble due to faulty installation or system kinks than there are genuine trap troubles. To prove this point we will list common "trap troubles" according to their symptoms.

In checking the performance of a suspected trap, the first thing to do is to break the outlet connection so that the discharge from the trap can be observed. If a test cock has been provided, this step is very simple. However, in far too many cases, no test valves have been installed and hence it is more or less of a job to break into the discharge line.

If inspection shows that the trap is not discharging, the trouble may be due to any of the following causes.

Pressure Too High

In any trap, a definite maximum force working direct or through a fixed leverage is available for pulling the valve from the seat against pressure. If the pressure against the valve seat is greater than the force available, the trap will not open. Failure of a reducing valve in the steam supply line is frequently responsible for the pressure building up. Faulty gages will often account for excessive steam pressure. Also, there is a possibility that traps with wrong sized orifices have been supplied by the dealer or larger sized orifices installed by the user but no change has been made in the pressure stamped or otherwise indicated on the trap.

There are many cases on record where the operating pressure in the plant may be raised without making any changes in the trap orifices. Assume that the plant pressure is raised from 100 to 125 lb. Let us further assume that the three-quarter inch traps for 100 lb. have the same valve seats that the manufacturer supplies in 125-lb. traps while the 1¼-in. traps for 125 lbs. have smaller orifices than in the 100-lb. traps. As a result, when the pressure is raised, the 1¼-in. traps will not open while the ¾-in. traps perform perfectly. Immediately the 1¼-in. traps are condemned as being "no good." If all the traps had stopped working, the cause

By T. H. Rea

would be more obvious but with some working and others balking, the reason is not so apparent.

Frequent seating of the conical valve will gradually enlarge the seating area, especially when soft valve materials are used. Manufacturers allow plenty of excess operating pressure in most cases but when unusually good service is obtained from these valve parts, they finally have to be replaced just because the seating area becomes too large.

No Water Coming to Trap

Broken siphon pipes, as cited previously, come under this classification. When dirt conditions are bad, elbows in the inlet line may become plugged. If such a condition is suspected, the trap should be taken off the line and the discharge watched. If no water comes, the trap certainly cannot be at fault. If several units are connected to a single trap, one or more of the units may become short circuited due to variation in pressure drop through the different machines or coils. This should not be charged against the trap as it is not provided with any apparatus to equalize pressure drops and to overcome the other causes for short circuiting. All the trap can do is pass water that enters the trap; this is a point that is too frequently overlooked. (See the previous article on p. 108 of the February issue.)

Trap May Be Filled With Dirt

The valves in float, upright bucket, and inverted bucket traps are opened when the float or bucket sinks. When the trap body is filled with dirt and scale, the bucket or float cannot sink and hence the valve does not open. This is a combination trap and system trouble, as a well-designed trap should remove a reasonable amount of dirt and only excessive amounts of scale, etc. are chargeable against the steam system itself.

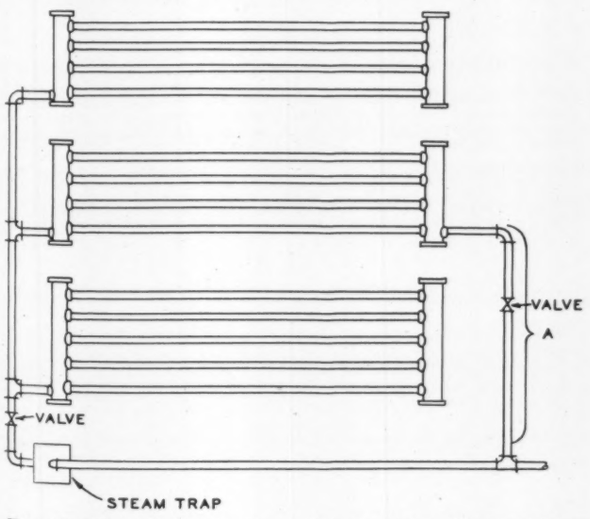


Fig. 22—The Trap in This Installation Would Blow Steam as Soon as the Steam Valve Was Opened Until the Line *A* Was Removed

Trap May Be Air-Bound

Unless positive means are employed to remove air, mechanically-operated traps will become air-bound. An obvious method for relieving air is to install an air cock. This is effective but causes a steam loss on account of the fact that steam and air both escape in the same proportion as they are found in the trap. The "non air-binding" upright bucket traps discharge a mixture of steam and air ahead of each condensate discharge. Hence these traps do not air bind unless considerable air has to be handled when very little condensate is coming to the trap. Air collects in the trap but cannot escape because there is not enough condensate to cause a discharge and thus relieve a portion of the air. In "inverted bucket" traps, air and steam escape through the bucket vent. The steam is condensed by the water that fills the body of the trap while air collects at the top of the trap. This air displaces water which is forced into the bucket until the bucket loses its buoyancy and sinks, thereby opening the valve and allowing the air to escape. Thus condensate is not necessary to have the trap pass air. However, inverted bucket traps will become air-bound if the bucket vent becomes plugged with dirt or grease.

Mechanical Condition

Inspection of the trap mechanism will quickly show whether or not failure of the trap to open is caused by mechanical trouble. The most common trouble of this sort is the valve or bucket being disconnected from the operating mechanism. Also, the valve may bind in the valve guide or distortion will cause enough friction to reduce the power of the mechanism and thus not be able to pull the valve away from the seat. The mechanism may wear loose or levers become bent so that float or bucket may be on the bottom of the trap and not exert any appreciable pull on the valve. In thermostatic traps the thermic unit may corrode, thus destroying its power to open the valve.

If, on testing the discharge line, it is found that the trap is discharging water continuously, causes of another sort are responsible for the condition.

The Trap May Be Too Small

A trap may have been installed that was simply too small for the job, either through ignorance of the amount of water to be handled or ignorance as to the capacity of the trap. Another possibility is that the right sized trap was purchased but fitted with a small orifice for a much higher pressure.

System trouble may be responsible for a trap discharging continuously. If water is carried over from the boiler, the traps may be overloaded. This condition will call for work on the boiler end or, if the priming is unavoidable, the installation of suitable purifiers or separators drained by emergency traps. Continuous discharge may also indicate a leak in the steam line so that water or other liquid is flowing into the coil and out through the trap.

The back pressure on the trap may be raised for some reason, such as other traps blowing through. This will reduce pressure differential through the trap and cut the trap capacity below the amount of condensate that has to be handled. This is a hard trouble to locate, as when

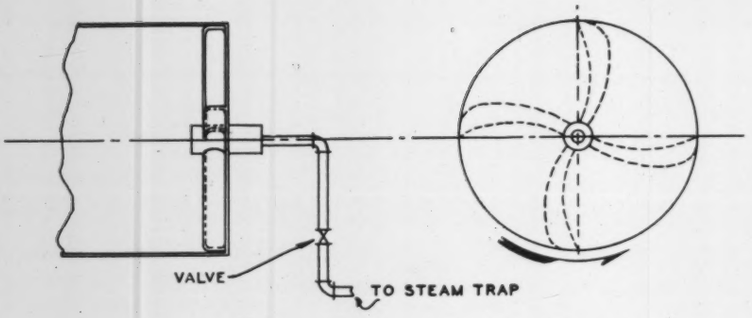


FIG. 23—BUCKET DRAINED CYLINDER

the trap discharge is broken for inspection the trap will have the proper pressure differential necessary for draining the unit.

Testing the discharge line may reveal that the trap is blowing steam as well as water. The chief reason for this is defective valves and seats. A leak through the discharge valve may be started by failure to lap the parts together properly when the trap was assembled, or the valve may not strike the seat squarely. Perhaps a fine piece of scale will lodge between the valve and the seat. If the valve does not open and close quickly, the fine particles of water carried at high velocity will cut or wire draw the valve metal. Once a leak is started, it quickly grows and unless the valve and seat are again lapped together, the valve or seat will be cut beyond repair. This means not only the expense of new parts but also a considerable steam loss during the time the valves were cutting out.

That a steam trap is no better than its valve and seat is self-evident. Durability of valve parts can be increased by three methods. First, the trap must be designed to open quickly and close quickly. This reduces wire-drawing to a minimum; it also insures tight seating and any pieces of scale lodging in the seat will quite likely be crushed, thus eliminating trouble from that source. The second way to increase the life of valve parts is to provide an adequate water seal. Steam will pass through a small crevice at a much higher velocity than water. Hence with a water-sealed valve, the seepage through a very small opening will be at low velocity and the cutting action of water alone will be negligible. However, if the small opening is not water sealed, the high velocity steam carrying minute particles of water will quickly enlarge the crevice and soon the valve is cut out. The third method is to use metals having high resistance to wire drawing as well as great hardness and tensile strength. If the parts are harder than ordinary pipe scale, pieces of scale cannot start small leaks by scratching the valve or seat. Likewise, the hard seat will not mushroom as quickly and lower the operating pressure.

Dirt or Scale May Prevent Valve from Closing

Regardless of the quality of the valve and seat material, large pieces of scale may prevent the valve from closing. Even though the valve parts are not damaged by this, condensate may escape through the partially blocked orifice faster than it is entering the trap, with the result that the trap soon blows steam. With such extreme scale conditions, strainers or scale pockets should be used.

Faulty Mechanism

A water-logged float or leaky bucket will keep the valve open and allow the trap to blow through. Improper adjustment of a thermic trap may prevent the valve from closing, especially at low temperatures and pressures. The remedy as with the mechanical trouble is new parts to put the trap in first class condition.

Loss of Prime Through Sudden Pressure Drops

This subject was covered in the article that appeared in the March issue of HEATING, PIPING AND AIR CONDITIONING. The remedy for this condition is to install suitable check valves on the inlet side of the trap.

If the steam trap operates satisfactorily with the discharge connection broken, but starts to balk as soon as it is connected with the return system, the pressure in the return lines is evidently too high. This may be caused by dirt or scale in the return line, or by another trap blowing steam. If the returns are brought to a vented receiver, the high pressure may be caused by a plugged vent. Sometimes vents in the boiler return trap will become closed, allowing pressure to build up.

The above does not cover all possible system and trap troubles. If a complete list could be compiled, it would soon be out of date as new kinks are continually being uncovered. However, we have endeavored to show that steam traps are not always at fault and that most types of trouble are easily located.

Before satisfactory service can be expected from a trap the following conditions must be fulfilled.

1. Trap must have sufficient capacity.
2. Trap must be in good mechanical condition.
3. The inlet line must be clear so water can get to the trap.
4. The discharge line must be open so excessive back pressure will not build up.

However, remember that a trap is an automatic machine and as such is entitled to periodic inspection and repair, as is any other piece of machinery.

Welded Steam Piping

In the erection of welded steam lines, the following factors are taken into consideration:

1. Pressure of the fluid.
2. Stress due to resistance to expansion of the piping system.
3. Stress due to weight of the piping system.
4. Internal stress in the metal from fabrication.

Combinations of these stresses in couples in high pressure steam piping may easily be sufficient to cause rupture.

The problem therefore requires:

1. Rigid welding control specifications.
2. Design of proper supports and anchors to preclude abnormal stresses.
3. Suitable arrangement to not only make the connections but provide the necessary flexibility in the space available.

"Open for Discussion"

A department in which we follow the custom of technical societies of allotting space in their programs for discussion of the papers presented

CHIMNEYS FOR TALL BUILDINGS

"Sizing Chimneys for Tall Buildings," *by Robert E. Hatlis.* Page 464, October, 1929, issue

SOMETHING very valuable has been developed in this article and it is quite true that the chimney in the ordinary high building is a matter of reduction rather than increase in size. This problem comes up again and again, but previous to the article in question the writer knows of no attempt to work out the problem in a labor-saving manner or by means of reduction curves such as are given therein.

Paragraph 2

On the other hand, there are some things in the article which seem to have been passed over with too little information for the average reader; no doubt the author, due to his familiarity with the subject, had no thought that such matters would not be immediately clear; but, one starting in to read this article without previous preparation has some difficulty in following out the discussion and in using the charts without a little more explanation, and it is the object of this letter to bring out that extra information so that the full value of the article may be available.

Paragraph 3

In Chart No. 1, for instance, on the right hand and left hand scales the notation is placed, "Diameter of the Stack in Inches." This should be plain enough for every one, but in running up these scales from the bottom the diameters in inches read as follows: 10-15-2-25-3-4-5-6-7-8-9-10-15-20-25-30, and so on by tens up to 100 at the top. There are two lines, both of which read 10, two which read 15 and another two which read 25. How can both of these lines be for the same diameter of stack when the diagonal line which they intersect is a fixed quantity line for so many 1,000 sq. ft. of radiation?

Paragraph 4

Closer inspection of the chart indicates that it is laid out on logarithmic paper so that the numbers on the right and left hand scales might be used the same as the numbers on a slide rule—that is, with any number of ciphers added or deducted and the product modified by having the same number of ciphers also added or deducted. But the objection to that is that the "thousands of sq. ft. of radiation" is not a direct product of the stack diameter, and, consequently, if the stack diameter is multiplied by 10 the radiation does *not* multiply by ten.

Paragraph 5

Clearer explanation of this chart is essential to intelligent use by the average person as well as a statement of the value of the top portion of the chart, say above the 30-30 line, into which the diagonal line does not extend.

Paragraph 6

In Chart No. 2 it is not clear why the height, *L,* is taken from the breeching and not from the grate level. In calculations for draft, it is customary to use the grate level in comparison with the top of the chimney and it can be seen that in a high boiler room, say 17 ft., or so, the chimney height would be at least 15 ft. more by taking the breeching level instead of the grate level.

Paragraph 7

This point also suggests the additional question as to why the outside temperature of 32 F is used. The article distinctly states, "As we are limiting our discussion to chimneys for low pressure boiler heating plants * * * " So the time of maximum boiler load and chimney load would be at the minimum outside temperature and not at 32 F, except in portions of the country where very mild winters are present. Would it not have been much better to take a zero outside temperature?

Paragraph 8

While the article also points out the undesirability of making chimneys too large in order to play safe, the method used in the article still produces a chimney apparently 100 per cent in excess of the actual requirements, as will be seen when it is considered that one of the biggest boiler manufacturers in the country starts off with a boiler rating 50 per cent in excess of the actual equivalent direct radiation load, this 50 per cent being divided into 25 per cent for pipe losses and 25 per cent for overload capacity in heating up in the morning. Therefore, if the boiler capacity is used in determining the chimney size (as every engineer will agree it should be) there is at once a 25 per cent overload capacity right there. Then if the chimney is made to carry a boiler overload of 50 per cent, this gives 125 per cent plus 50 per cent of 125 per cent, or 187½ per cent. Added to this is the matter of using an outside temperature of 32 deg. instead of zero, which adds another 7 per cent of the 187½ per cent, or 13½ per cent more, making just 200 per cent. Yet the actual load on the chimney can never go over 125 per cent of the boiler rating even when heating up in the morning.

Paragraph 9

No mention is made in the article of unlined steel stacks, and yet there are a great many in use; in fact, it is rather unusual to find a purely heating chimney lined all the way up to the top. Actually, this is unnecessary, as the gases from a heating boiler seldom get over 500 degrees, and are nearly always discharged directly into an unlined steel flue about ⅛ in., or 3/16 in. thick through which they travel anywhere from 20 ft., to 100 ft., to the chimney and radiate some of their heat

Solution for example quoted in Mr. Hattis' article as computed by Mr. Alt:

Example: Size of chimney for two — 22,000 sq. ft. rad. boilers on stack 250 ft. high.

Two — 22,000 sq. ft. rad. boilers = 44,000 sq. ft. rad. $\times \frac{240}{33300} = 317$ hp.

Based on 5 lb. of coal per hp. = 1,585 lb. coal and 7.5 lb. air per second per combustion.

Draft loss in fire at 10 lb. coal per sq. ft. of grate per hr. and using No. 3 buckwheat	0.40 inches
Draft loss in boiler (fire tube at 100 per cent rating)	0.20 inches
50 ft. of breeching at 0.10 in. per C ft.	0.05 inches
Three 90 deg. turns in gases at 0.05 inches each	0.15 inches
Contingencies	0.10 inches
Total draft loss excluding chimney losses	0.90 inches
Theoretical draft in chimney with 500° gases and zero outside at sea level about 0.90 in. per C ft.; for 250 ft. height	2.25 inches
Deduction of draft losses outside of chimney	0.90 inches
Draft loss which may occur in chimney	1.35 inches
Divided into 250 ft. length gives per C ft	0.54 inches
At sea level a masonry chimney will carry 7.5 lb. of gases per second with a friction loss of 0.50 in. per C ft. of height if diameter is	30 inches
A steel stack under like conditions must be	28 inches

If 50 per cent overload capacity is to be included the pounds of gas per second will increase from 7.5 to 11.2 and the chimney diameters will then become 36 in. and 34 in. respectively.

on the way. Why should a lining then be placed in the chimney after the gases have been cooled by passing through an unlined steel breeching?

Paragraph 10

Still one other point which is not clear is how a proper intensity of draft is insured; some boilers require more and some less; some fuels require more and some less; the combustion rate also has a bearing on the matter; how is this taken care of? And how is a chimney in Denver sized from the charts? It could not be sized the same as one in New York owing to the greater elevation. The article speaks of the quantity *D*, but nowhere does it state what *D* represents; is it the stack diameter and is it in inches or feet?

Paragraph 11

If the author can produce curves to take care of the points suggested they would be of infinite value; or if the curves as given can be modified so as to take those points into consideration, the value of the curves would be greatly enhanced. No doubt, with the pioneering work which Mr. Hattis has already done along this line, the refinements suggested will not be difficult and he can be sure that the results will be as interesting as they are valuable.—*Harold L. Alt.*

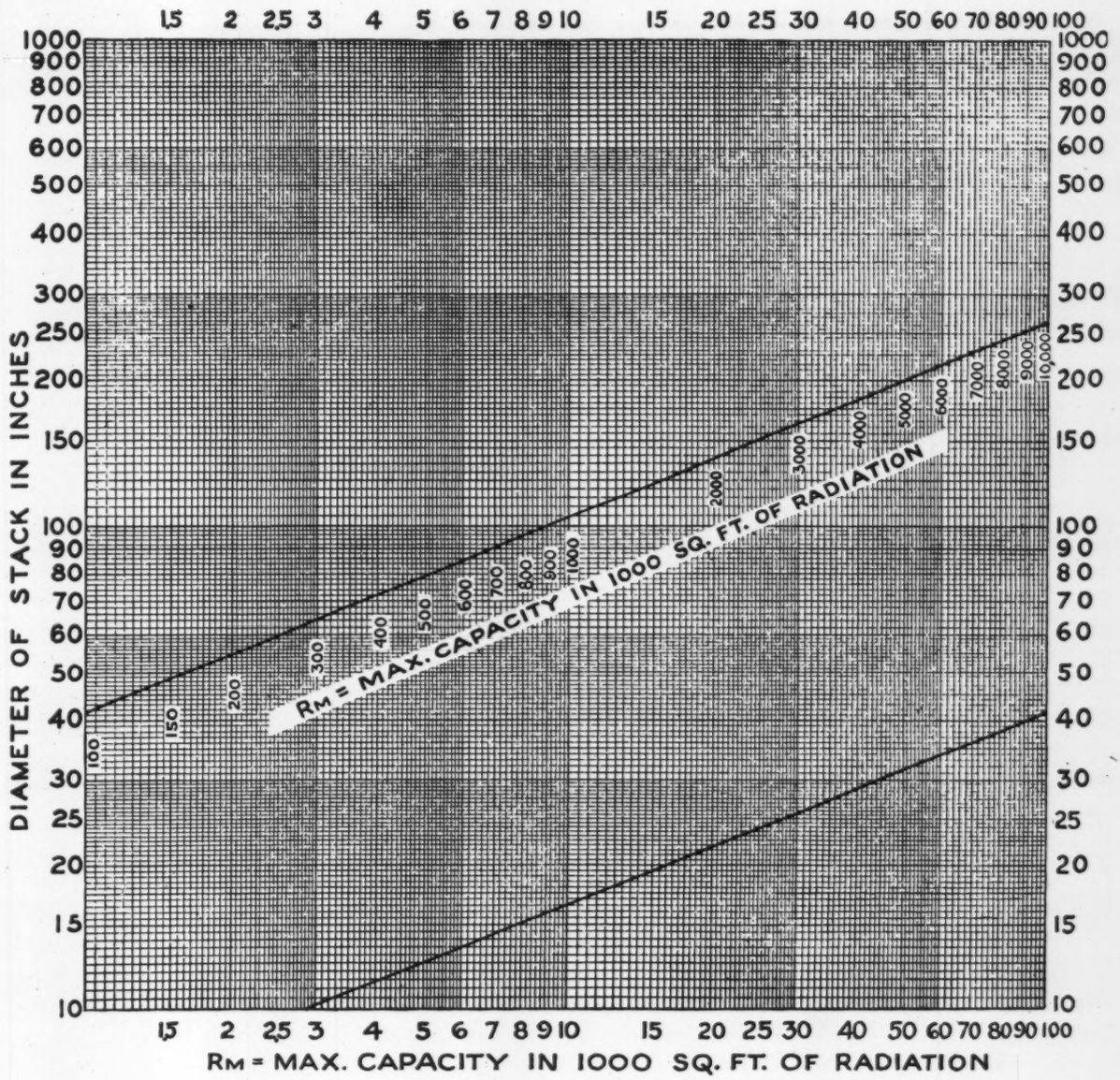
Reply of Mr. Hattis

In Mr. Alt's first paragraph, he mentions the thought which inspired the writing of this article, namely, a labor-saving method of determining how much to reduce boiler manufacturers' recommended stack diameters when the actual heights are greater than those recommended. This, and this alone, was the purpose of this article. It was not the intention to devise a new way of computing chimney sizes or to criticise the stack recommendations of boiler manufacturers.

In connection with Mr. Alt's second paragraph, he comments on the lack of explanation, etc., which he feels necessary for complete understanding of the subject. I have used reams of paper in arriving at the equations which are mentioned in the body of the article and illustrated in the charts. I have deliberately refrained from including the laborious reduction equations so as to give the results of this effort and describe its use. In fact, I have said: "Here is a problem," "Here are some charts," "Their equations are these," "They are based on these assumptions," "Here is how to use the charts to obtain these results, "I believe you will save time."

In Mr. Alt's third paragraph, he calls attention to an error that I also noticed in the figures for the ordinates of Chart No. 1. I made verbal comment to the editor of HEATING, PIPING AND AIR CONDITIONING, but delayed written comment pending a let-up in my business. All the comment of this paragraph will be cleared up by the following correct list of figures for these ordinates of Chart No. 1. Starting from the bottom up these should be: 10-15-20-25-30-40-50-60-70-80-90-100-150-200-250-300-400-500-600-700-800-900-1000. The curve does not extend beyond 260, as this figure represents the diameter of the stack in inches and provides for stacks as large in diameter as 21 feet. The correct figures for the lower abscissa from left to right are as follows: 1.5-2-2.5-3-4-5-6-7-8-9-10-15-20-25-30-40-50-60-70-80-90-100. (See the corrected chart on the next page.)

In Mr. Alt's fourth paragraph, he raises the question of the propriety of the use of logarithmic paper for representing Chart No. 1. On page 466 of the October issue is given the equation represented by this chart, which is as follows: $R_m = 9.17\ D^{5/2}$ where R_m = maxi-



CORRECTED CHART NO. 1

mum stack capacity in thousand square feet of radiation and D = diameter of stack in inches.

Does Mr. Alt question the propriety of representing this equation on logarithmic paper?

In the discussion above of Mr. Alt's third paragraph I believe I have answered Mr. Alt's fifth paragraph.

In Mr. Alt's sixth paragraph, he raises the question regarding my using L in Chart No. 2 as the height of the stack above the breeching. As it is my object in these charts to equate the available drafts of two stacks, I cannot complicate it by boiler and breeching losses. Suppose the stack height called for is 100 feet and the breeching is 12 feet above the grate, substitute for the low stack height 88 ft. and solve with this figure substituted. Then use the same breeching size as that recommended by the boiler man. A tapering reduction in breeching size at the stack juncture is desirable.

In Mr. Alt's seventh paragraph, he raises the question of the use of 32 F, recommending the use of zero degrees. As these charts are not intended for purposes of determining stack diameters from gas volume and friction losses, but purely to get the stack equivalent to that recommended by the boiler man, I decided to use 32 F as a base, with the idea of leaning a bit on the side of safety. The equation for Chart No. 2 on the basis of 32 F is:

$$D_s = .0076\ L$$

For zero F outside temperature this equation becomes $D_s = .0086\ L$. This 13 per cent excess may compensate for the ambitious 70 per cent boiler efficiency assumed.

In Mr. Alt's eighth paragraph, his comments are really criticisms of the stack recommendations of boiler manufacturers. From my observations in the field, with the various conditions of boiler firing, cleaning, maintenance, efficiency, etc., I feel that the recommendations of the boiler manufacturer take more account of the practical limitations encountered than theoretical but none-the-less correct figures of a competent engineer.

In Mr. Alt's ninth paragraph, he raises the question

of lining stacks. My charts have nothing in them referring to stack lining. We happen to have ordinances governing partial stack lining in Chicago and we generally cover or line our breechings. The question of linings was raised in discussing space required for inside stacks. Regarding the loss of temperature (also mentioned by Mr. Alt in this paragraph) in unlined steel stacks, that is a question affecting stack gas temperatures and is worthy of discussion in a separate article. Remember my article is not a treatise on chimneys.

In Mr. Alt's tenth paragraph, he raises several questions—proper intensity of draft, variation in friction loss through different fuels and fuel beds, variations due to altitude, and the apparent lack of explanation of exactly what the letter *D* used in my article represents. As my article was not a general treatise on chimneys, but merely a presentation of a method for easy calculation of the diameter of a tall stack equivalent to a short stack recommended by your boiler manufacturer, I feel that I am asked questions that are outside of the scope of my article. In any book on chimneys (J. C. Mingle's "Draft and Capacity of Chimneys" is a thorough treatise) can be found answers covering many phases of stack design. Regarding the explanation of the letter *D* on page 466 under the heading "Low Stack," in stating the problem, I say, "Given *L* in feet, *D* in inches," etc. For greater clearness I should have said that *D* represents the stack diameter.

In Mr. Alt's eleventh paragraph, he invites me to do a tremendous amount of additional work. May I say that now that I have started something in the way of labor-saving charts, will some other energetic engineer carry on? When I find time for more refinements, I shall be glad to contribute my bit.

In Mr. Alt's computation of the example given in my article, he arrives at a figure of 34 inches, whereas my article gives 33 inches. A long discussion can ensue from the questions raised by analysis of the figures used in Mr. Alt's article, but the answer he gets differs so little from mine that I feel more elated than ever about my slight contribution to the subject of stack calculations.—*Robert E. Hattis.*

A. S. H. V. E. Code of Minimum Requirements for Heating and Ventilation of Buildings

By Samuel R. Lewis

"The Code of Minimum Requirements," ***published by the American Society of Heating and Ventilating Engineers.***

SECTION IV: This section refers to minimum requirements for the determination of amount of indirect steam or hot water radiating surface to be installed in indirect gravity and fan circulation heating and ventilating systems.

The section sets forth fundamental data, much of which is given in the catalogs of the manufacturers, and is perhaps subject to criticism because it lists the proprietory types of certain manufacturers by name but cannot hope to cover all types by all manufacturers. Since the section was printed, several exceedingly efficient new types of indirect radiation have been developed. It is my opinion that in the next reprinting the names of manufacturers should be omitted.

Very valuable formulas and diagrams are given with examples, so that the amount of air and of heat to be handled, and the temperatures by indirect heating systems, can be calculated.

Complete data for designing gravity indirect heating and ventilating systems are given, though throughout the Code, as in most practical applications of gravity indirect heating, there is a tendency to ignore the item of *control.*

Where the apparatus serves one room only, and where the heaters are in several separately-valved sections, it is possible to achieve control of the delivery temperature with valves and thermostats, but under all other conditions recourse must be had to dampers.

Most gravity indirect heating systems are crude in design and cruder in operation, and neither valves nor dampers are used in controlling them. When dampers in bypasses around the heaters are used, it is important that the area for air going through the heaters shall be curtailed in proportion as the area for air going through the bypass is increased, and that the air going through the bypass shall not be allowed to touch either side of the heater or to "back-lash."

The arrangement shown in sketch *A* is very satisfactory when automatic temperature regulation is available.

The arrangement shown in sketch *B* is adapted to manual control and is often found to give better results than any automatic means for controlling tempering heaters.

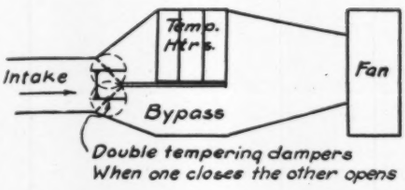
When mixing dampers are used with gravity indirect heaters there is usually a cessation of air flow when the warm air from the heater is shut off; cold air not being inclined to travel up flues. It is practical to use tempering heaters with gravity indirect systems, and there are many satisfactory plants, some of large size, in which this has been done.

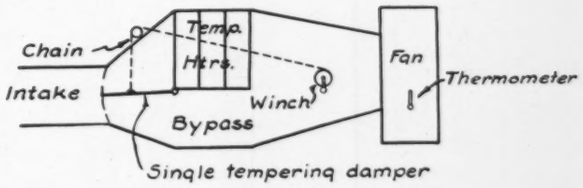
In school building gravity indirect systems it is usually helpful to have the bypass above the heater, rather than below it, as per sketch *C*. This really is only an adaptation of sketch *B* and can of course be used with a double-blade damper controlled by a thermostat, the same as in *A*.

The return piping from gravity indirect heating systems should be kept out of the cold air supply rooms, as it will pretty surely be frozen sooner or later if of the gravity wet return type.

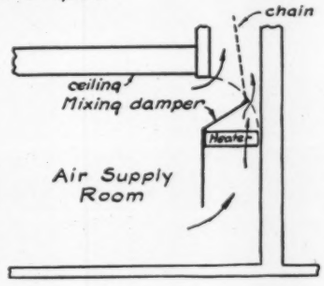
In last month's HEATING, PIPING AND AIR CONDITIONING, *Samuel R. Lewis discussed the first three sections of the American Society of Heating and Ventilating Engineers' Code of Minimum Requirements. Mr. Lewis takes up the fourth, fifth, sixth, and seventh sections of the Code this month; in future issues, he will comment on the other sections.*

Appearing with Mr. Lewis' comments in the March number was a brief description of the Code of Minimum Requirements.





SKETCH A (LEFT), B (RIGHT), AND C (BELOW)



The Code gives interesting data on aspirating flues, but who uses aspirating flues any more? There is no more wasteful method of carrying air than to heat chimneys for that purpose. Fans are much less wasteful.

The Code is not happy in its items which deal with registers. No one seems to know what a register really is. The trade practice is full of contradictions. The following definitions, for which I stand sponsor, are synthetic ones, gleaned from the catalogs of several leading manufacturers and tacitly are satisfactory to the majority of register manufacturers.

Register: A register is a metal covering of open design attached to movable valves for controlling, in the room, the flow of warm or cold air.

The term register includes the face and the body containing the movable valves and operating movement complete.

Grille: A grille is a register face with wheel slot and frame hole openings closed up in order to make the design continuous.

Register Face: A register face is the exposed front of the register. It has a wheel slot and frame hole openings for use when attaching it to the register body.

Border: A border is a frame set in the floor into which the register fits, used principally as a fire protection.

Register Body: A register body consists of the valves and body frame which holds the valves. A register body is a register less the face.

Body Frame: A frame into which the valves are set and to which the register face is attached.

Valve: A louvre or damper which may be opened or closed to admit or to prevent the circulation of air.

Pulley Register: A register with pulleys attached so that, when the operating handle is out of reach, the register may be opened and closed by cords or chains.

Wall Frame: A four-sided reverse beveled frame of cast iron usually masoned in brick walls to which a register, register face, or grille is attached.

Size: Sizes are opening sizes, i.e., the size of the opening to be covered.

I think the Code should have used the term "Grilles," instead of "Registers."

Section V: This concerns minimum capacity and installation requirements for Low Pressure Steam and Hot Water Boilers.

This laconic section is admirable from every standpoint. I wish that I had written it.

It will be noted that "square feet of radiation" are touched very delicately, so exceedingly delicately that they are hard to find.

Nothing is said about the boiler *efficiency*. It is difficult in a logical Code to say anything about efficiency. This matter of efficiency is a factor of the combustion, and combustion is being accomplished more and more in special apparatus not furnished by the boiler manufacturer. Few manufacturers of heating boilers today can state what percentage of the heat their boiler will extract, when supplied with a given volume of carbon-free products of combustion at a given temperature. I predict, however, that the boiler maker of the future will be compelled to state just this, and that the amount of heat absorbing surface will have very much to do with the answer, just as it does today in power boilers.

The future manufacturer of stokers, gas burners, oil burners, etc., will not hesitate—does not today hesitate—to state the rate of efficiency with which his device will convert the fuel into clear gaseous products of given temperature.

Looking at the matter in this fashion we may eliminate or track to its lair the mysterious element claimed by certain manufacturers of cast iron boilers, which entitles them, due to some peculiarity of construction, to an alleged heat absorption per unit of area far beyond that known in steel boiler practice, and which allows them an alleged efficiency in the high figures despite a gas outlet temperature of 1,000 to 1,200 degree.

Section VI: This part relates to minimum requirements for the design and installation of warm air furnace heating plants.

The Code sets forth rational formulas by which the basic computations for design of warm air furnace heating may be made on a heat unit basis. It gives excellent specifications for the construction of an assuredly satisfactory system.

It will be noted from the drawings which accompany the Code that the air supply duct drops to the floor far enough away from the furnace to prevent radiant heat from the furnace warming the down-goer enough to set up reverse currents.

It may be noted also that while asbestos paper is specified as an additional precaution against fire when warm air ducts come close to it, no asbestos paper is specified for the outside of free-running bright tin ducts. Asbestos paper on bright tin free-running ducts has been found to increase rather than to retard transfer of heat from the warm air inside the ducts to the cooler air outside of the ducts.

It must be remembered that the flow of air in gravity warm air heating systems depends on temperature-differences, the useful head which causes the air-flow being governed by well-understood natural laws. When long

horizontal runs are necessary, better results will be obtained by a riser close to the furnace which will certainly get warm, followed by a horizontal run with a favorable slope, than with the riser at a more remote point preceded by the horizontal run. It is as important that the cool air in the room to be heated shall have opportunity to fall out of the room, as that the warm air from the furnace shall be given a channel through which to flow into the room.

The Code refers entirely to *gravity* warm air heating though it fails so to state. Doubtless this will be corrected in future issues.

Section VII: This section of the Code pertains to minimum requirements for the design and installation of chimneys or stacks for steam boilers, hot water boilers and warm air furnaces.

This is an excellent code for computing chimney sizes and for chimney construction.

It requires that all chimneys shall be tested with a smudge fire for leakage. Anyone who has tested the average chimney will testify to the presence of frequent and surprising leaks. In many cases when the top of the chimney is closed, smoke will appear around window frames many feet distant from the chimney.

The tile flue lining is a frequent offender, as it is usually difficult to point up the butt joints tightly between the superimposed sections of the lining. Unless the chimney is perfectly straight, the joints in the lining cannot fit perfectly against each other, and if the lining is small in cross-sectional internal area, the mason cannot reach down inside to strike off the mortar properly. There is also a human tendency, when a tile lining is used, to depend too much on it, not filling full all joints between the surrounding brick with mortar. The appearance of smoke at remote points is a sure evidence that the mason has merely buttered the edges of the abutting brick or tile pieces, leaving hollow channels which give free air passage in the interior of the otherwise tight-appearing wall.

In general, it is evident that a chimney built with struck joints all slushed full of mortar gives promise of being a better chimney than one built carelessly, even though the latter has first quality unbroken and unchipped tile linings.

The Code does not mention a few kinks which it might be of interest to record here:

A chimney cannot very well serve two heaters if one only burns gas. Gas burning heaters do not require any particular draft. They have draft hoods open into the heater room to prevent possible back drafts which might blow out pilot flames, and so when the companion heater needs a strong suction, as with coal, the gas heater breeching acts as a draft-check. In such situations there should be separate chimneys.

The relation of the chimney to the roof is of great importance, since wind eddies whipping over the higher parts of the roof will cause down drafts unless the chimney top is above or remote from their influence.

The ornamental corbeling out and tapering back of the chimney top, often the pride of the old time masons, has an important practical value, just as has a bell top on a metal chimney, since the sloping up to the outlet from the projection tends to divert strong horizontal air currents upward at an angle, creating an injector-like effect and promoting the draft.

The table of chimney sizes would be improved in my judgment if heat units instead of "square feet" were substituted. Some perplexity on the part of the user might be avoided since now, having discounted the expanded rating in square feet imposed by the heater manufacturer, the layman is perplexed as to which "square foot" rating he should use in picking the chimney from the table.

It would simplify matters also for the warm air furnace designer and would remove a taint of discrimination if the simpler heat unit basis were used for all heaters.—*S. R. Lewis.*

Cooling by Evaporation

In a locality, for instance, where the wet bulb temperature is 78 degrees and the dry bulb temperature 90 degrees, the wet bulb depression (the difference between the wet and dry bulb) would be 12 degrees. In another city the dry bulb temperature is 95 degrees with wet bulb at 78 degrees with a wet bulb depression of 17 degrees. A greater cooling can be accomplished in the latter instance, and the greater the wet bulb depression the greater the possibility of cooling.

Toledo Museum of Art Heated by Forced Circulation Hot Water and Mechanically-Ventilated
Edward B. Green and Sons, Albert Haut Hopkins, Architects; Samuel R. Lewis,
Consulting Engineer

Heating·Piping
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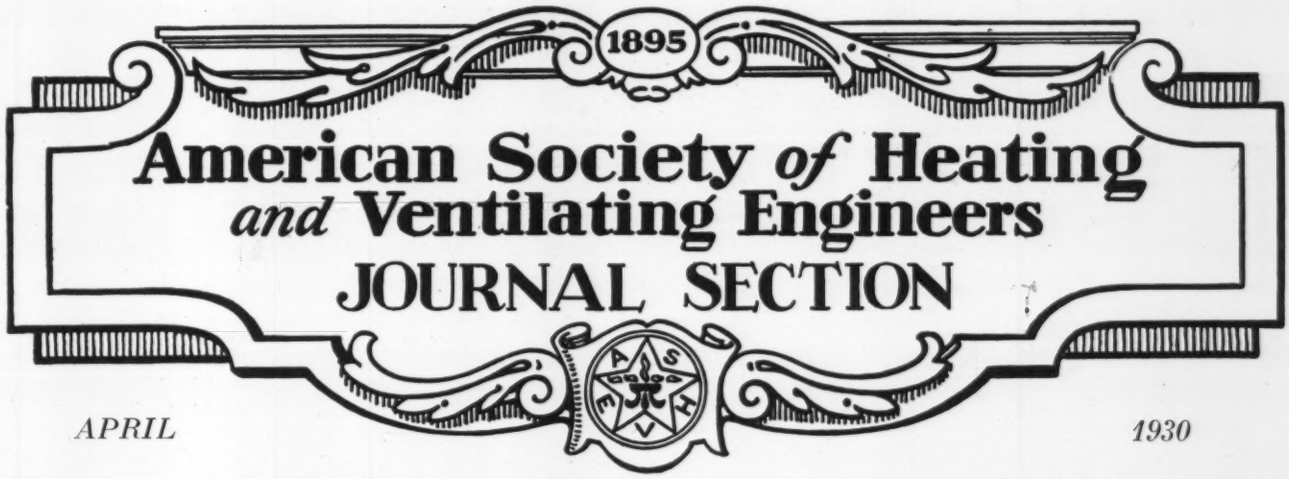

APRIL *1930*

Material in the Section prepared by Publication Committee: G. L. Larson, *Chairman*; H. H. Angus, and W. A. Rowe ; A. V. Hutchinson, Manager of Publications

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Development of a Method for Heat Regulation

By F. I. Raymond (*MEMBER*) and R. D. Lambert (*NON-MEMBER*), Chicago, Ill.

THE purpose of this paper is to describe and explain an apparatus for and a method of automatically controlling the temperature in steam heated buildings. Automatic control should provide a comfortable temperature in all parts of the building at all times, and should reduce operating costs. The apparatus to be described in this paper does this, *first,* by providing a heat control mechanism and, *second,* by providing a method of operation which makes possible an even distribution of steam at reduced heat capacities.

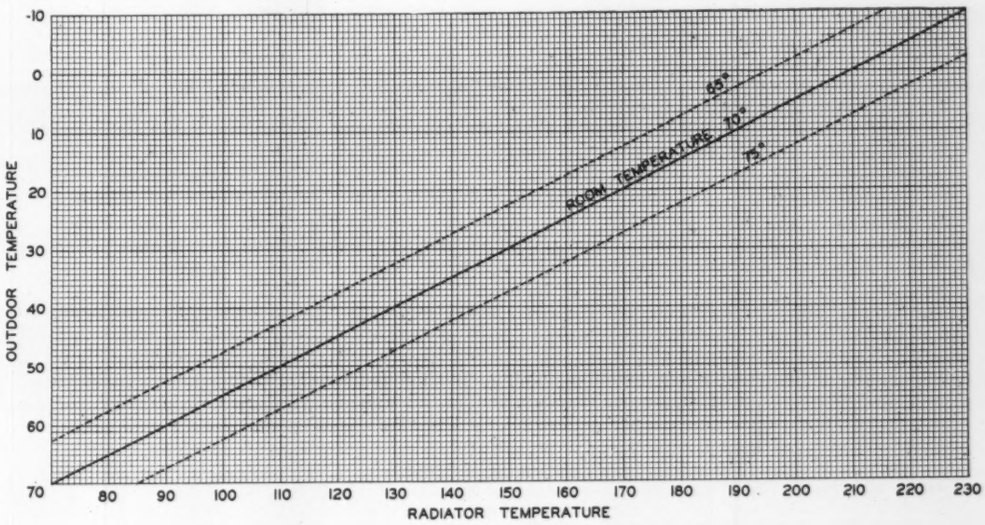

FIG. 1—CURVES SHOWING RELATION BETWEEN OUTDOOR TEMPERATURE, ROOM TEMPERATURE AND RADIATOR TEMPERATURE

The design of this apparatus is based on fundamentals of heat control going beyond any of the common methods now in use. For this reason its design will be reviewed before describing the apparatus and its operation. The scope of this paper makes it impossible to go as far into the theoretical considerations of temperature control as would be desirable. Many of the statements made would have to be qualified to fit all the situations which might be encountered. These cases can best be handled in discussion.

To maintain a comfortable temperature in a building the considerations are:

1. Heat emission proportional to heat loss.
2. Even distribution of heat.
3. Avoidance of *overshooting* and *lag.*

It should be the purpose of an adequate temperature control apparatus to accomplish these results, but in order to design such an apparatus, the basic means of accomplishing each of these functions must be ascertained. The ruggedness of the apparatus and simplicity of operation are additional factors to be taken into consideration in the design.

The room temperature varies directly as the heat emission of the radiators and inversely as the heat loss of the room. For practical purposes the heat emission of a radiator standing in air at 70 F. varies directly with the radiator temperature, and the heat loss from a room at 70 F. varies inversely with the outdoor temperature. These relations are not strictly true as the increments of heat emission for each degree of temperature difference between the radiator and surrounding air increase as the temperature difference increases. But since this is also true of the heat losses of the room the two errors tend to offset each other.

There are other considerations besides outdoor temperature which affect the heat loss, such as wind and sun. However, this is one case in which the human element, usually far from helpful, actually aids in the solution of the problem. On the windward side of the building the occupants will keep the windows closed, relying on infiltration for ventilation. On the leeward side, especially if it is in the sun, the occupants are apt to have the windows open for ventilation. In this way the factors of wind and sun are somewhat minimized and an outdoor thermometer, protected from the sun, gives a fairly accurate indication of the heat loss. From this it is apparent that the room temperature can be considered to be a function of the radiator temperature and the outdoor temperature. This relation is shown in Fig. 1.

The control of heat emission from the radiator can be effected by (1) varying the steam pressure, (2) fractionally filling the radiator or (3) combinations of these

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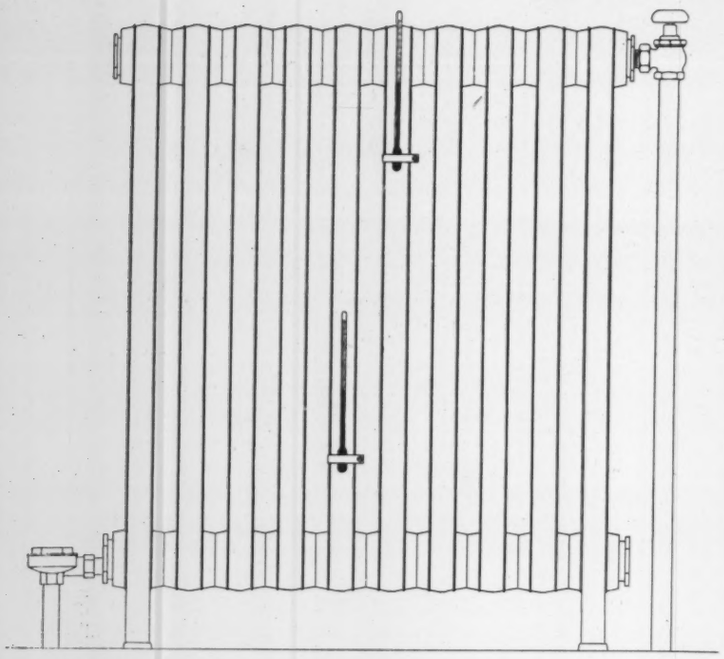

FIG. 2—TYPICAL ARRANGEMENT OF THERMOMETERS FOR METHODS (2) AND (3)

two methods. However, the average temperature of the radiator metal is the only true indicator of the heat which will be emitted by a radiator standing in a room having a temperature of 70 F. For method (1) a single thermometer in intimate contact with the radiator surface will be sufficient to indicate the heat emission at various steam pressures, but this method is limited to a temperature range of about 250 F. to 130 F. (15 lb. pressure to 25 in. vacuum). For methods (2) and (3), two or more thermometers are necessary to give an average temperature of the full and empty portions of a fractionally filled radiator. (See Fig. 2.)

Since method (1) *i. e.,* varying the steam pressure, has the limitation of a 250 F. to 130 F. radiator temperature, it is necessary that an adequate temperature regulation system utilize fractionally filled radiators to maintain the correct relationship between radiator temperature and outdoor temperature shown in Fig. 1. A radiator temperature of 130 F. is equivalent to a 40 F. outdoor temperature. During the greater part of the heating season the outdoor temperature in many localities is above 40 F. and a regulation system relying only on vacuum to secure the correct radiator temperature would be ineffectual during this part.

The correct relation between heat loss and heat emission in any one room is of little value unless this relationship is maintained in all parts of the building. It is very important that the radiation be designed to fit the heat losses and the piping to give even distribution of heat. But even with the heating system balanced for one pressure condition, the distribution may not be equal at all other pressures. Therefore, a temperature regulation system should operate so as to bring about even distribution of heat under all conditions.

This could best be accomplished by keeping the steam lines full at all times and, by means of a controlled valve at each radiator, admitting steam to the radiator as required to keep the radiator at the correct temperature. This method is impractical because of expense. The same results can be obtained by keeping the mains full of steam at the same pressure as in the radiators and, by short bursts of additional pressure, forcing steam into the radiators as required. By this means the inequalities of piping design are somewhat evened out, and in a fairly well designed system, approximately equal amounts of steam are forced into all radiators under the various conditions of pressure and vacuum.

The adequate temperature control system should eliminate the objectionable conditions of *overshooting* and *lag*. These conditions are bound to occur unless the relation between radiator temperature and outdoor temperature shown in Fig. 1 is maintained at least approximately. *Overshooting* occurs when a radiator, emitting heat at a greater rate than the heat loss, is shut off as the room reaches the correct temperature but, continuing to emit heat at the high rate for a period, raises the room temperature excessively. *Lag* occurs when this radiator is turned on again as the room drops to the correct temperature but, since the room cools from the outside wall *in* and since the usual place for measuring room temperature is an inside wall, the room becomes uncomfortably cool near the outside walls before the temperature falls below that required on the inside wall.

It is plain that intermittent heating will produce some *overshooting* and *lag*. If the periods are short and frequent, they will be reduced to a minimum. If the periods are long and infrequent, such as exist in a system controlled by a room thermostat, they are apt to produce uncomfortable temperatures. This will be especially true during the moderate portions of the heating season. However, if frequent periods of intermittent steam supply are used, the relationship indicated in Fig. 1 is approximately maintained, and the *overshooting* and *lag* are negligible.

With these principles of temperature regulation established, the control apparatus and its operation on a

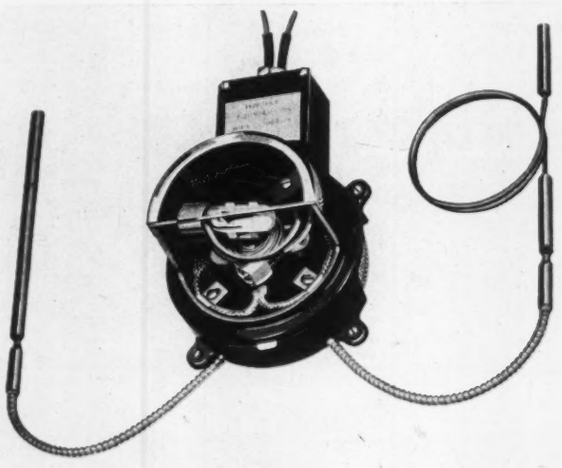

FIG. 3—CUT-AWAY VIEW OF CONTROL INSTRUMENTS

simple steam heating installation will be described to illustrate how the design and operation comply with these principles. Fig. 3 shows a cutaway view of the control instrument. The instrument consists of a case in which is mounted a mercury switch and a Bourdon tube. The Bourdon tube operates the mercury switch and thereby controls the oil burner, coal stoker, gas burner or, in the case of central steam service and zoned heating systems, the steam valve. The Bourdon tube is operated by the expansion and contraction in three metal bulbs which are attached to the Bourdon tube by flexible capillary tubing. One of the metal bulbs is placed outdoors and the other two are attached to a suitable radiator.

It is the total expansion of the liquid in the three bulbs which determines the action of the Bourdon tube and thereby the mercury switch. When the outdoor bulb is warm, the liquid in this bulb will be expanded and just a slight amount of heat applied to the radiator bulbs will be sufficient to move the mercury switch to the off position. When the outdoor bulb is cold, the liquid will be contracted so that it will be necessary that the radiator bulbs be heated to a high temperature in order to move the mercury switch to the *off* position. The instrument will therefore cause the heating unit to operate in a manner which will produce the desired temperature in the radiator to which its bulb is attached.

For the standard steam heating system the instrument is furnished with an outdoor bulb having twice the volume of the radiator bulbs. Thus each degree drop in outdoor temperature produces a 2-deg. increase in the radiator temperature. The instrument is provided with 20 ft. of capillary tubing on the radiator bulb and with 10 ft. of tubing on the outdoor bulb. A recess is provided in the back of the instrument to hold any excess of tubing over the length required. In Fig. 3 the tubing is shown coiled up in this recess. The instrument is usually installed in a closet or hallway close to the control radiator with its flexible tubes leading outdoors and to the radiator. Electric wires connect the instrument to the automatic control for the heating plant. Fig. 4 shows a typical installation of the instrument.

In a steam, vacuum, or vapor heating system the radiator bulbs of the instrument are always applied to the last radiator on the last riser on the longest steam main. The reason for using this radiator for the control is to insure a steam supply to all the radiators or at least to the risers leading to all the radiators before the last

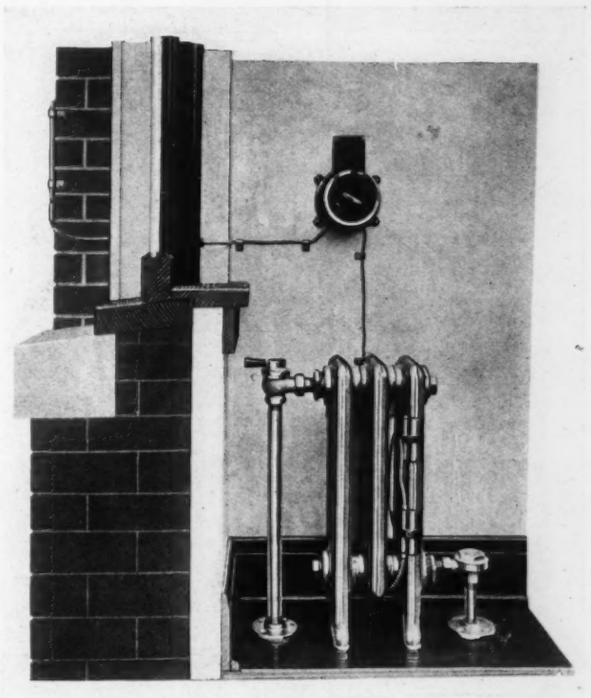

FIG. 4—TYPICAL INSTALLATION OF CONTROL INSTRUMENTS

radiator can be heated sufficiently to shut off the supply of steam.

In order to illustrate the operation of the apparatus, the various cycles of operation in a typical heating plant will be considered. When the instrument starts the steam supply (See Fig. 5A), steam is forced into the piping and the radiators, finally reaching the control radiator on the end of the longest run of pipe. Steam enters the radiator until the radiator has been brought to the correct average temperature, when the instrument stops the steam supply (See Fig. 5B).

During this cycle the steam main has been completely filled with steam, since steam must reach the last radiator before the control operates, but the amount of steam entering the radiators may be unequal, depending on how well the piping is balanced and on the condition of the air valves.

As the control radiator condenses steam and its average temperature drops below the correct level, the con-

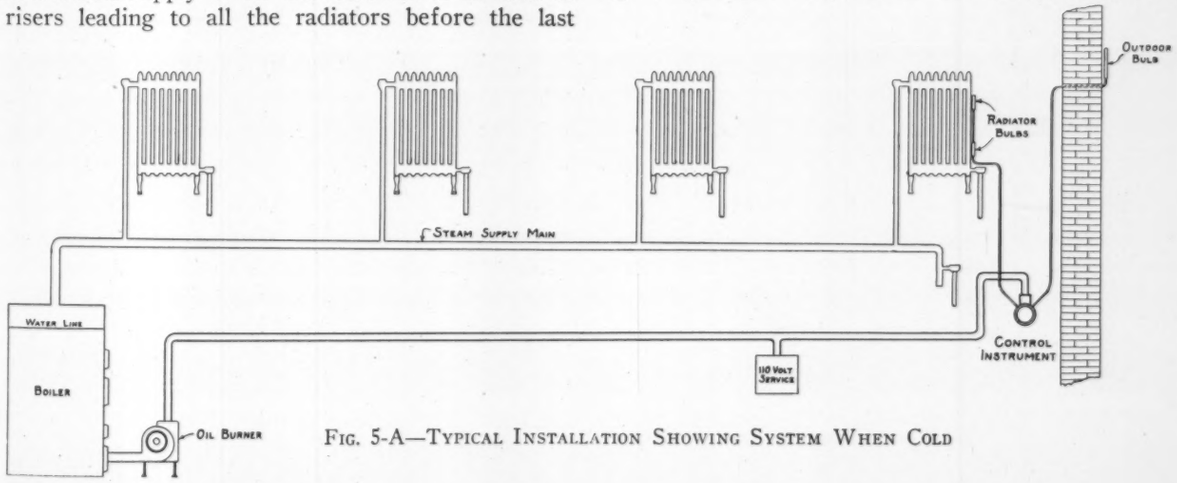

FIG. 5-A—TYPICAL INSTALLATION SHOWING SYSTEM WHEN COLD

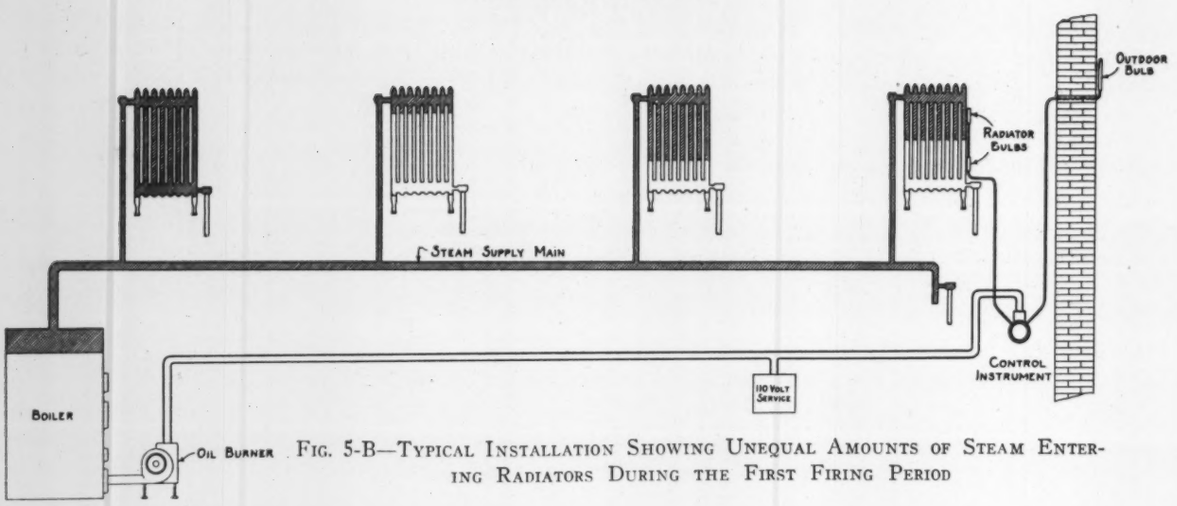

FIG. 5-B—TYPICAL INSTALLATION SHOWING UNEQUAL AMOUNTS OF STEAM ENTERING RADIATORS DURING THE FIRST FIRING PERIOD

trol apparatus starts generation of steam. Meanwhile steam has been condensing in the other radiators at a rate proportional to the surface in contact with steam, so that the least filled radiator condenses the least steam. Steam has also been condensing in the mains, but with covered pipes, the simmering of the boiler at off periods will be sufficient to keep them full of steam (Fig. 5C).

As the control apparatus starts the generation of steam for the second time, with the mains full of steam, the increase in pressure is much more even at each radiator, forcing approximately equal amounts of steam into each one (Fig. 5D). Thus with the radiators containing the least steam tending to condense less and the bursts of pressure at frequent intervals tending to send equal amounts of steam into each radiator, there will be a decided tendency for the steam to equalize in the radiators after a few cycles (Fig. 5E).

As before mentioned, the three considerations in the design of an adequate temperature control system are (1) heat emission proportional to heat loss, (2) even distribution of heat and (3) avoidance of *overshooting* and *lag*. The instrument must be rugged and its operation simple.

The apparatus described in this paper provides for the first consideration with the counter-balancing action of the outdoor bulb and the radiator bulbs acting on the same Bourdon spring. The problem is solved by going to the real bases, *i. e.*, radiator temperature and outdoor temperature. The second consideration is taken care of by the cycles in the operation of the system, as shown in Figs. 5A to 5E. These cycles tend to bring about more even distribution of steam with each operation, even when the system is not properly balanced. Of course, a balanced system is imperative for the most satisfactory operation. However, the balancing effect of the cycles is always acting at all pressures and vacuums.

The third consideration is taken care of automatically in the solution of the first two. With radiator temperatures held to values equivalent to outdoor temperatures there can be no overheating or underheating. And with the short frequent cycles necessitated by this apparatus the possibility of *overshooting* and *lag* are entirely eliminated.

With this explanation of the design and description of the instrument and its operation, its application to different systems of heating commonly used in large and small buildings can now be described. A modified form of this apparatus is being used for hot water heating systems, which is beyond the scope of this paper. The application to one-pipe steam systems, as used in apartments and commercial buildings, and to vacuum systems, as used in the larger apartments and office buildings, will now be explained. It will be shown how this apparatus operates in conjunction with other control systems, especially with the zone control system, using a specially designed gradual opening valve.

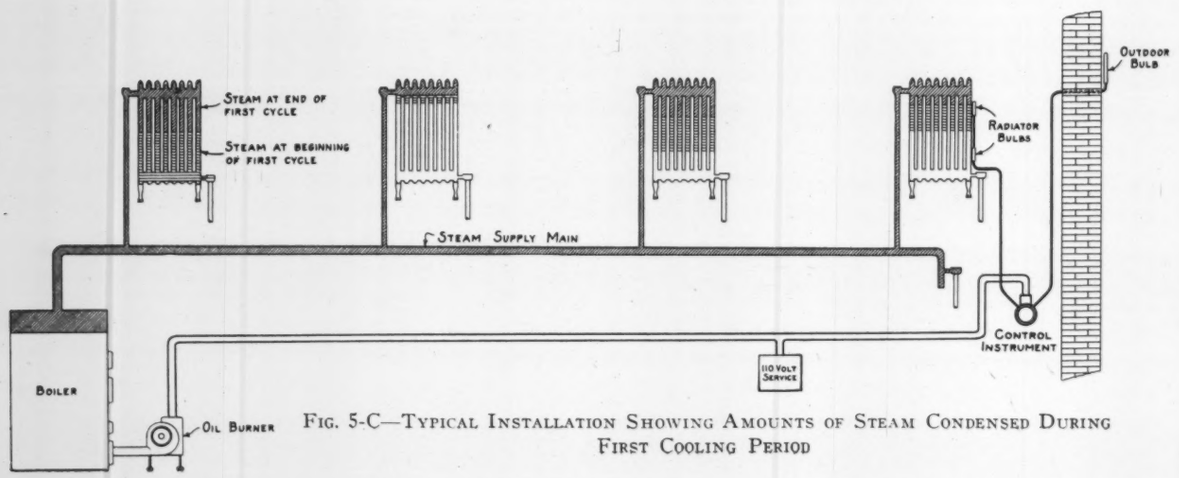

FIG. 5-C—TYPICAL INSTALLATION SHOWING AMOUNTS OF STEAM CONDENSED DURING FIRST COOLING PERIOD

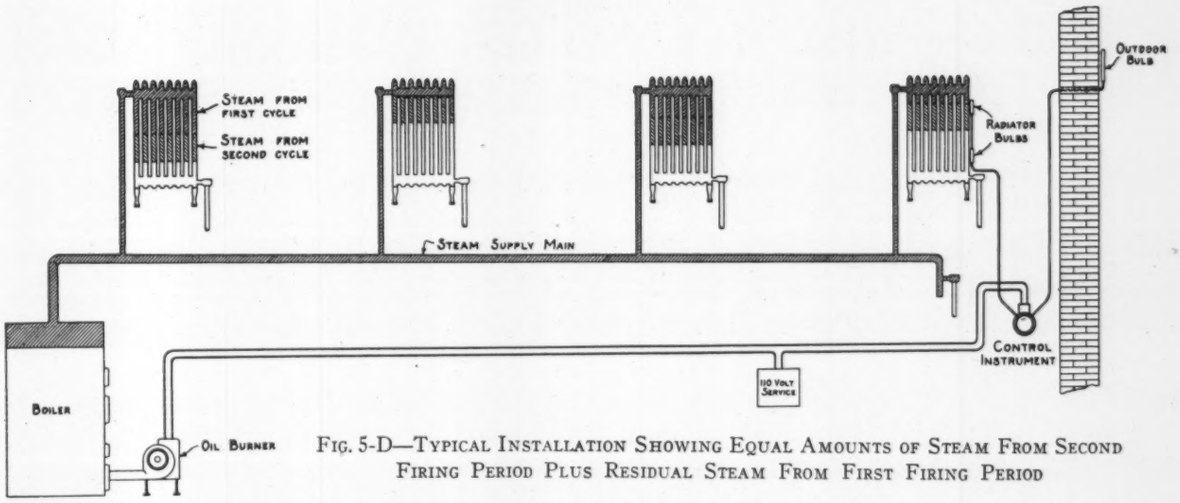

Fig. 5-D—Typical Installation Showing Equal Amounts of Steam From Second Firing Period Plus Residual Steam From First Firing Period

The operation of the one-pipe or two-pipe steam systems controlled by the apparatus is shown in Figs. 5A to 5E. The operation remains the same if there are two or more mains and risers. The control radiator is always the last radiator on the longest run of pipe. If this is on the top floor it is advisable to place the outdoor control bulb on a short mast on the roof. Otherwise it is placed on the outside wall, protected from the sun in all cases.

One-pipe vacuum systems and the various types of two-pipe vacuum systems, with or without pumps, are used to secure better distribution or a more moderate temperature range at the radiator. Thermostatic room control, the maintenance of a constant differential and such devices as modulating valves, return traps, and orifices in the supply line give added refinement in operation. With all of these, this apparatus affords the primary control.

This apparatus is valuable, when used with thermostatic room control, in that it eliminates the *overheating* and *lag*. The thermostat will determine when heat is needed, and the apparatus will determine the rate at which heat is to be delivered. Under this combined control there will be a long and moderated delivery of heat to the room each time the thermostat calls for heat rather than a short and intense delivery. If gradual acting thermostats are used the apparatus is of value as a master control to prevent excessive delivery of heat when tenants throw the thermostats out of operation by opening windows or by abusive manipulation of the thermostats.

In conjunction with the two-pipe vacuum system, the apparatus functions to prevent excessive differential during the moderate part of the heating season. This condition is objectionable because it makes modulating valves, orifices, etc., inoperative. The excess of differential over frictional resistance is caused by the closing of the return traps on the radiators. With fractional radiator heating afforded by the apparatus, the return traps will remain open until cold weather requires a full radiator when steam pressure can be built up. With the return traps open, the full effect of the return pump is available for circulation. It is seen that with these systems, the apparatus is the primary temperature control, aided and refined by the other devices.

The zone system of heating is becoming popular because it provides a means of controlling the heat input to various parts of the building to meet the variations in the conditions of exposure. This zone system necessitates the use of valves, either manually or motor controlled, to adjust the flow of steam to each zone, and some method of control for these valves. A gradual-acting, motor-operated valve has been developed to operate for zone control in conjunction with the apparatus described in this paper. The valve is shown in Fig. 6. This valve is operated by two motors, one to close and one to open,

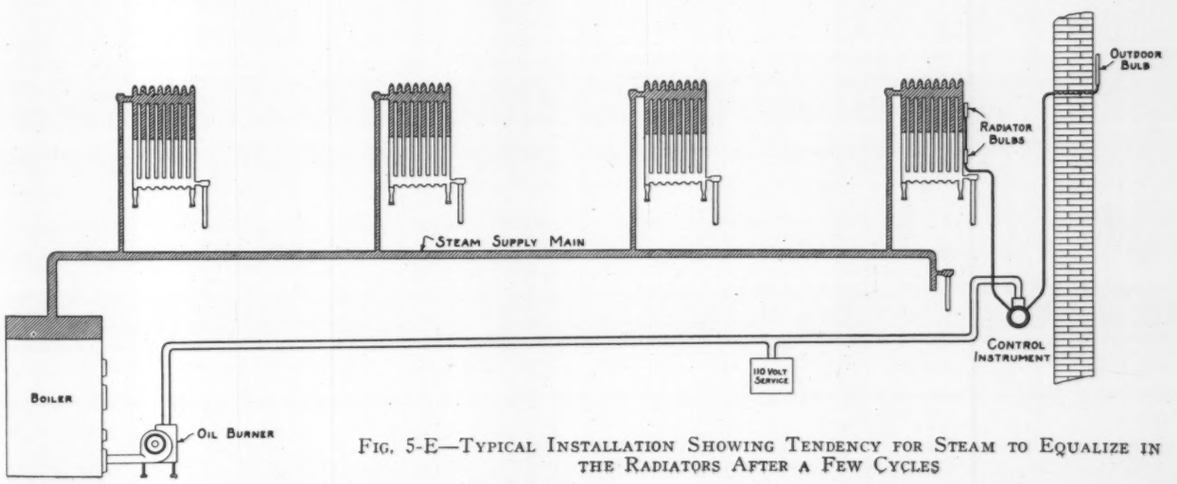

Fig. 5-E—Typical Installation Showing Tendency for Steam to Equalize in the Radiators After a Few Cycles

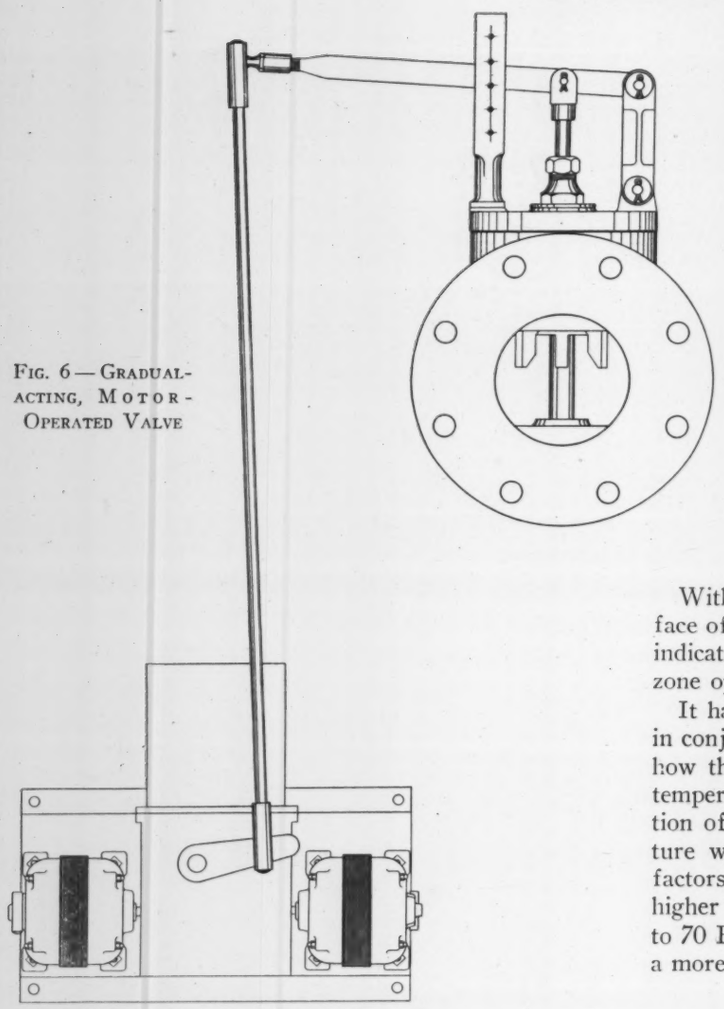

FIG. 6 — GRADUAL-ACTING, MOTOR-OPERATED VALVE

and the time required from full open to full closed, and from full closed to full open is twenty minutes.

Fig. 7 shows a heating system divided into four zones, one for each face of the building. Steam is supplied to each zone through a gradual-acting, motor-operated valve. Each valve is controlled by the apparatus connected to the control radiator in each zone. This is the radiator farthest from the supply valve. The valves can also be operated from a switchboard in the boiler room for night operation and morning pick-up.

Each zone, from supply valve to control radiator, acts in practically the same way as the simple system shown in Figs. 5A to 5E. Due to the long closing and opening time of the motor-operated valves, the changes in temperature of the control radiator will occur before the valve is completely closed or opened. This will cause the valve to operate from one fractional position to another rather than from full open to full closed. The steam supply will therefore be continuous under a slightly oscillating pressure.

The outdoor bulb for each zone is placed on the face of the building which that zone heats. The bulb is protected from direct sunshine by a shield but is open to circulation of the *skin* air of the building. High wind will wash the face of the building thereby producing a lower *skin* or surface temperature. On that face of the building where there is no wind there will be a rising current of air along the face of the building due to the heating effect of the building. This produces a higher *skin* temperature. Similarly, sunshine on one face of the building will produce a higher *skin* temperature. Thus the heat supplied to each zone is controlled by the *skin* temperature of the building in that zone, which automatically compensates for the variables of wind and sunshine, as well as for outdoor temperature.

With this system of zone control, the rooms on each face of the building are supplied with heat in the amount indicated by the *skin* temperature on that face. Each zone operates as a complete, independent unit.

It has been shown by the description of its operation in conjunction with the various types of heating systems how this control apparatus will maintain a comfortable temperature in all parts of a building. By the elimination of *overshooting* and *lag*, this comfortable temperature will be lower than in the presence of these two factors since a room which has been at a temperature higher than 70 F. is no longer comfortable as it cools to 70 F. Therefore, a fuel saving is obtained as well as a more equable temperature.

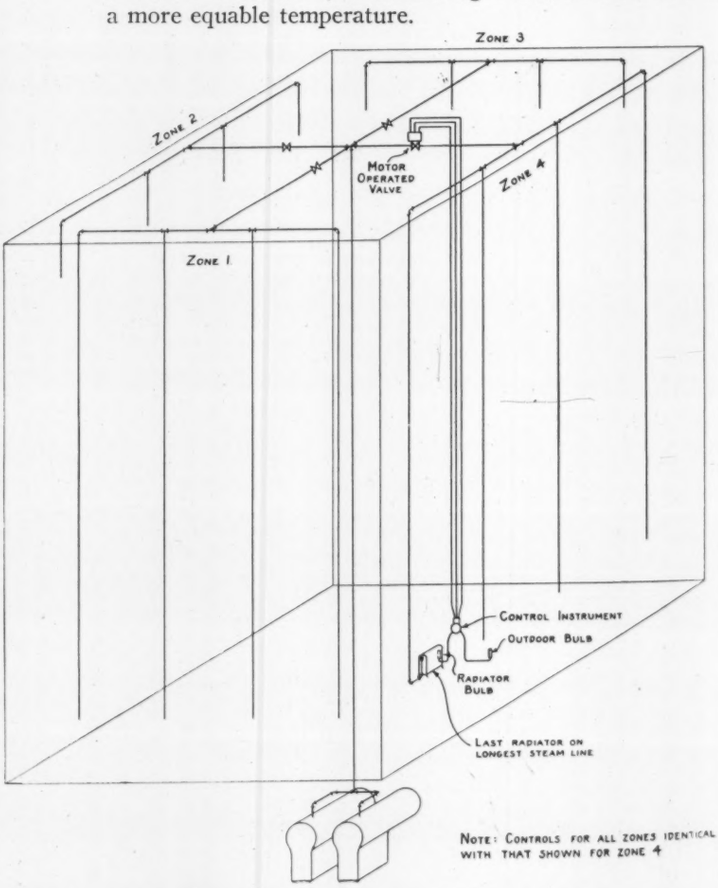

FIG. 7—HEATING SYSTEM DIVIDED INTO FOUR ZONES, ONE FOR EACH FACE OF THE BUILDING

FIG. 1—WARM-AIR HEATING RESEARCH RESIDENCE AT THE UNIVERSITY OF ILLINOIS

Wall Surface Temperatures

By A. C. Willard[1] and A. P. Kratz,[2] Urbana, Illinois
MEMBERS

A great deal of attention has been devoted in recent years to the so-called surface coefficients for building walls, and especially to the ratio which exists between the outside and inside surface coefficients, which must be known in order to compute the overall heat transmission coefficient for any given wall.

It so happens that this ratio is always the reciprocal of the ratio of the temperature drops at the surfaces at the inside and outside faces of the wall, and hence may be accurately ascertained in the case of any existing wall by simply measuring the inside and outside air temperatures and the corresponding surface temperatures.

Such measurements have been in progress at the University of Illinois for several years, and a study of the results reveals the fact that an entirely new significance attaches to the inside surface temperature of a building wall, and the difference between this temperature and the air temperature inside the building. The inside wall surface temperature not only affects personal comfort, but may be made use of for estimating the relative thermal values of various walls as actually built.

For example, would you be more comfortable on a zero day in (1) a building with outside walls made entirely of thin sheet iron, or (2) a building with outside walls made entirely of thick corkboard, provided *the same "breathing line" temperature of* 70 *F existed in both buildings?*

In other words, the thermal effectiveness of the walls used in a heated or cooled building is reflected in the difference in temperature which that wall maintains between the inside air temperature at the "breathing line" and the inside wall surface temperature at the same level. This paper discusses the new significance of such temperatures and presents actual original data on the subject.

Introduction

During the investigation[3] of various types of heating systems at the *University of Illinois* an increasing amount of attention has been given to the surface temperatures of the exposed walls of heated rooms as actually determined under typical winter weather conditions. In the past, much importance has been attached to the air temperatures at various levels within the heated rooms, and such temperatures, usually taken at the "breathing line" 5 feet above the floor, have been regarded as a measure of heating plant performance. Any plant which maintained 70 F at the breathing line in coldest weather was supposed to satisfy the heating guarantees and the occupant.

The occupant has not always agreed with this idea. In the first place, he occupies a zone at an average height

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[3] An investigation of warm air furnace heating has been in progress at the *University of Illinois* since October, 1918, under a co-operative research agreement between the *National Warm Air Heating Association* and the University. Six Engineering Experiment Station Bulletins and one Circular have already been published.

An investigation of direct systems of steam and hot-water heating has also been in progress since April, 1926, under a similar co-operative research agreement between the *Institute of Boiler and Radiator Manufacturers*, the *Illinois Master Plumbers' Association*, and the University. Two Engineering Experiment Station Bulletins have already been published.

ILLINOIS ENGINEERING EXPERIMENT STATION

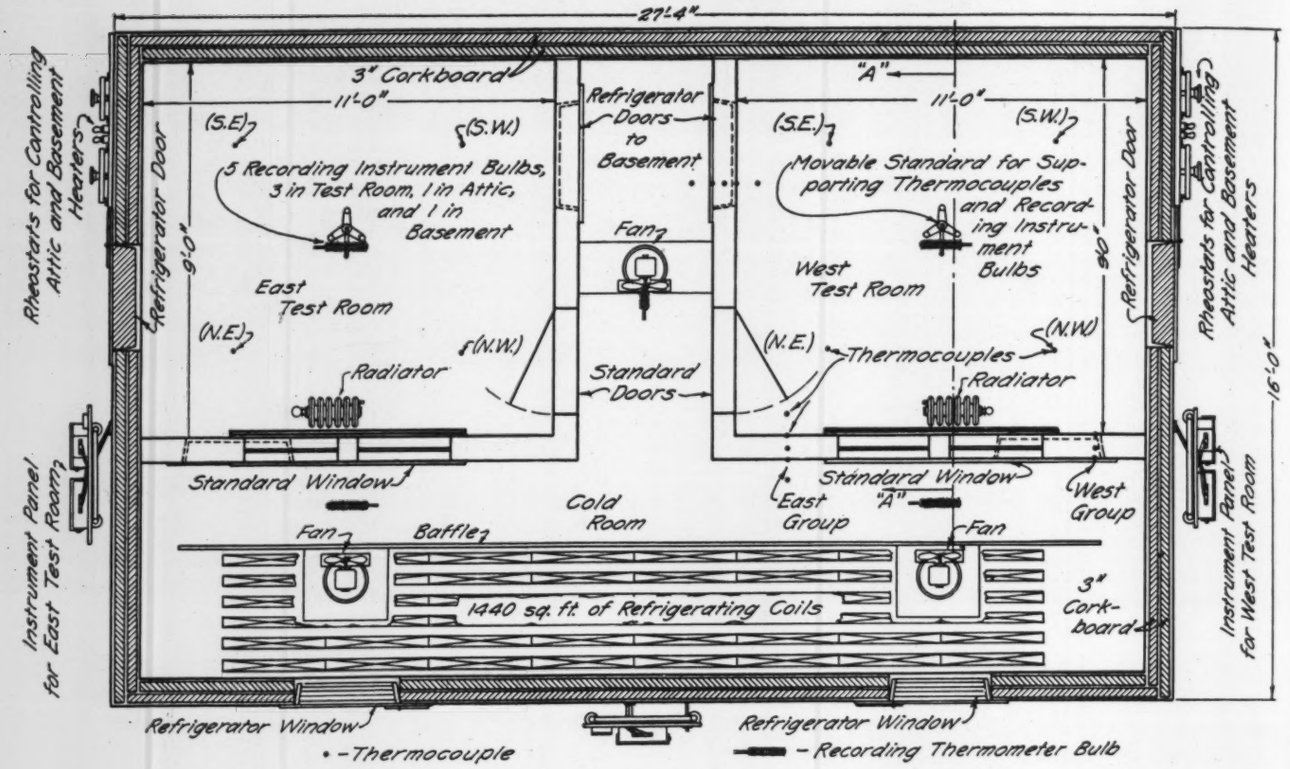


FIG. 2—DIRECT-STEAM AND HOT WATER HEATING LABORATORY PLANT AT THE UNIVERSITY OF ILLINOIS

of 2 feet 6 inches above the floor where the air is always colder[4] than at the breathing line. How much colder depends on the type of heating plant and heating unit and house construction, but is not the subject of the present discussion, important though it is. In the second place, but possibly of more importance than the first item, the occupant is also subjected to the radiation effect of the cold surfaces of the exposed walls and glass (assuming floors and ceilings are next to heated spaces), *which are always colder than the air in the room at the same level.*

There is nothing particularly novel about the preceding statement, but the heating engineer appears to have taken very little interest in this fact aside from the academic reference made to it in setting up the general equations for heat flow through any wall. The surface temperature on the inner, or room side, of the exposed wall is usually denoted as (t_1), and, "since it is unknown," is promptly eliminated from the final equation and from the reader's mind. Unfortunately, the occupant in the actual room is keenly conscious of its reality, and the colder the weather and the poorer the wall construction, the more conscious of his cold surroundings he becomes *even though all "breathing line" temperatures are exactly 70 F.*

Summary of Results

Actual measurements, over a considerable period of time, of both air and wall surface temperatures, at the Research Residence (Fig. 1) and at the room heating testing plant (Fig. 2) in the Laboratory at Urbana, with the breathing line temperature at approximately 70 F in all cases, show:

(1) A rapid increase in the difference between air temperatures at the "breathing line" and the inner wall surface temperatures at the same level, as the outside air temperature drops.

Reference to Fig. 3 will show that when the indoor-outdoor air temperature difference increases from 20 to 70 F, the difference between the temperature of the air at the breathing line and the inner wall surface increases from 2.5 to 12.5 F, although there is no change in the indoor air temperature at the breathing line. If the occupants were just comfortable in the first case when the outdoor air temperature was approximately 50 F, they will not be comfortable in the second case when the outdoor air temperature was approximately 0 F.

(2) For any given outside air temperature, the inside air to surface difference in temperature increases with the outside wind velocity over the wall.

The very marked effect of wind velocity on the "inner air to surface" difference in temperature is strikingly shown in Figs. 4 and 6 which present data from the room heating testing plant in the Laboratory. Note the drop and change in slope of the inside wall surface temperature curve caused by wind (curve No. 3 compared with curve No. 2).

(3) At any outside air temperature, the "inner air to surface" difference in temperature is greater than the "outer surface to air" difference in temperature, except for still air conditions.

An inspection of Fig. 3 at any indoor-outdoor air temperature difference shows the temperature difference A is always greater than B.

(4) For a wall exposed to actual weather conditions with wind velocities ranging from 5 to 10 miles per hour,

[4] University of Illinois Engineering Experiment Station Bulletins No. 189, pages 60 and 61, 67 and 68, and No. 192, pages 40 to 44.

the ratio between the "inner air to surface" difference and the "outer surface to air" difference in temperature increases rapidly as the outside temperatures drop.

This fact has never been recognized or allowed for in heat loss calculations for moderate temperature differences. The decided increase in this ratio occurring at the Research Residence is shown in Fig. 3, from which it may be noted that the "inner air to surface" difference in temperature A is not only greater than the "outer air to surface" difference B, but the ratio A/B also increases from 1.30 to 1.83, with the same wind velocity in all cases. Reference to Fig. 6, however, for the Laboratory plant indicates that at approximately the same wind velocity, the ratio A/B is practically constant at a value of 3.2 over the whole range of outdoor temperatures.

A number of factors are present in the case of the wall at the Research Residence that are not present in the Laboratory plant. The principal difference in the two cases is that the wall in the Research Residence is always under the influence of some solar radiation, or "sky-shine," even on dark and cloudy days. This effect is completely lacking in the case of the wall in the Laboratory plant. An explanation of the difference in the behavior of the two walls can be based on the hypothesis that this difference is caused entirely by solar radiation. If the latter is present, the outside surface of the wall will be warmed and the ratio A/B will be correspondingly reduced. Furthermore, if the radiation is more intense, or the ratio of sunshine to no sunshine is greater on mild days than on colder ones, the ratio A/B will be reduced more in mild weather than in cold. This is exactly the condition shown in Fig. 3.

In the case of the Laboratory plant (Fig. 6), where the sun or solar radiation effect is absent, there is no warming effect to be correlated with the milder temperatures, and a constant value of the ratio A/B over the whole temperature range is to be expected. Since there is no warming effect from solar radiation, it is also evident that the actual value of the ratio will be higher than for the case in which the solar radiation is present. These hypotheses are also supported by the results shown in Fig. 6.

The conditions at the Laboratory plant, with the total absence of solar radiation, are similar to the laboratory conditions under which the ratios commonly used in practice were determined, and it is of some interest to note that the usual assumption of a ratio of approximately 3 to 1 at about 15 miles per hour wind velocity corresponds very closely with the ratio of 3.2 to 1 found in the Laboratory plant. Furthermore, even in the case of the Research Residence, where the solar radiation somewhat modifies the ratio, the value of 1.83 to 1 obtained at 70 F indoor-outdoor difference and 10 mile wind velocity, when corrected to a 15 mile wind velocity, is reasonably close, for all practical purposes, to the ratio of 3 to 1 usually assumed. The fact that such modification occurs, however, should always be taken into consideration where walls exposed to actual weather conditions are concerned.

(5) The ratio between the "inner air to surface" difference and the "outer surface to air" difference in temperature increases rapidly at any given outside air temperature as the outside wind velocity increases. A fact which is recognized and allowed for in heat loss calculations by making the outside surface coefficient greater than the inside surface coefficient. For an inside-outside air temperature difference of 70 F with 15 mile wind velocity, the ratio is ordinarily assumed to be approximately 3:1, ranging from 2.5:1 to 3.5:1. Note especially that the 3:1 ratio does not apply precisely for 15 mile wind velocity at an inside-outside air temperature difference much less than 70 F, since the ratio is affected by the inside-outside air temperature difference as well as the outside wind velocity.

Reference to Figs. 4 and 6 from the Laboratory plant will show that the ratio of "inner air to surface" temperature difference to "outer surface to air" temperature difference increases from approximately $11.6/11.5 = 1.00$ for still air to $17.5/5.5 = 3.18$ for a wind velocity of approximately 10 miles per hour.

Attention is again directed to the absence of solar radiation or "sky-shine" in this plant which operates to reduce somewhat the temperature of the outside wall surface below that for an actual house, and hence the ratio of 3.18 at 10 miles wind velocity is larger than for the same wind velocity at the Research Residence.

(6) Walls exposed to sun effect, especially south walls, show marked reduction in "inner air to surface" differences in temperature (improving personal comfort) for many hours during, and for some time after periods of sunshine. See the period from 7:30 a. m. to 10:00 p. m. on January 7, 1929, Fig. 5, and note that the sun effect lasted long after sunset and darkness, although the

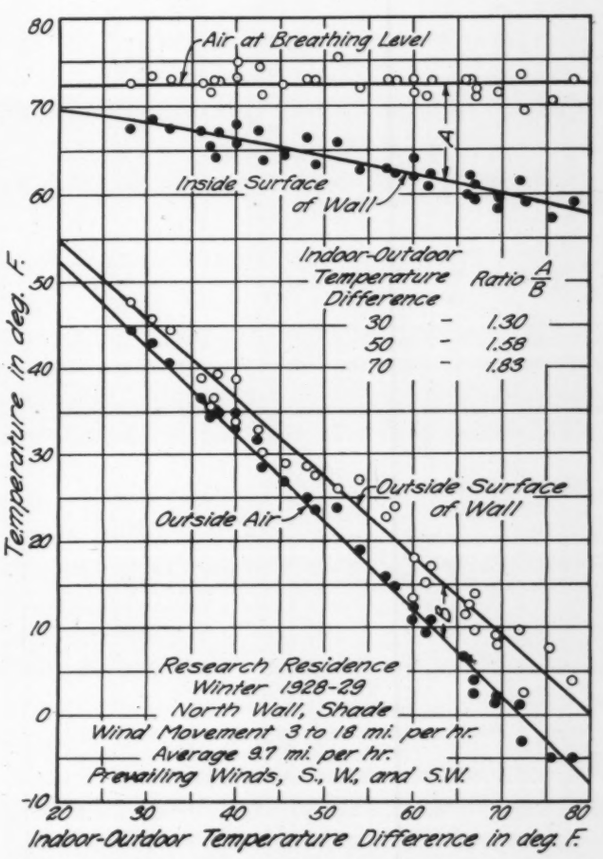


Fig. 3—Inside and Outside Air and Wall Surface Temperatures at Research Residence

outside air temperatures ranged only from 3 F to + 7 F during this period.

The "outside wall to air" differences in temperature show tremendous *increases* for the same south wall during periods of sunshine, ranging from a difference of 8 F at 7:30 a. m. to 43 F at 11:00 a. m. on January 7, 1929, Fig. 5.

(7) On cloudy days, there is apparently very little distinction between the "inner air to surface" difference in temperature of north and south walls. The same statement applies almost equally well to the "outer surface to air" difference in temperature. See Fig. 5, excepting the period from 7:30 a. m. to 3:30 p. m. on January 6, and the period from 7:30 a. m. to 10:00 p. m. on January 7, 1929, when sun effect was more or less active.

It is noteworthy, however, that the outside surface temperature of the south wall was always slightly higher than the outside surface temperature of the north wall even with no sunshine and at night.

Conclusions and a Suggested Application

(1) Entirely too little attention has been given to the radiation effect of the inside surfaces of exposed wall and glass on the personal comfort of the occupants of heated rooms, especially during "coldest weather."

(2) The use of air temperatures at the "breathing line" alone, as the sole index of satisfactory thermal conditions for room occupancy, is crude and may result in much dissatisfaction on the part of the occupants, since they are subjected to much colder air temperatures at the "comfort line" 2 feet 6 inches above the floor, and also to the radiation effect of much colder exposed wall and glass surfaces at all levels in the room.

(3) The present ratio of 3:1 for the outside to inside surface coefficients used in the academic calculations of heat transmission coefficients of solid walls and glass is only valid for an inside air to outside air temperature difference of approximately 70 F and a wind velocity of approximately 15 miles per hour. *Both air temperature difference* and wind velocity affect this ratio.

(4) Unquestionably, a distinction should be made between north and south rooms having exactly the same *calculated* heat loss, and the same wall and glass exposure, in selecting the sizes of heating units or capacities for such rooms. The distinction should be accomplished by an *addition* to the heating equipment of the north room, and not by a reduction for the south room.

Temperature control or regulation, preferably automatic, is far more important in south rooms than in north rooms, as attempts to control an entire building from the main unit *cannot be successful on sunshiny days.*

(5) The effective insulation of buildings assumes an entirely new significance when the differences between the inside air and inside wall or glass surface temperatures for various types of wall construction and glazing are considered and compared. Effective wall and glass insulation accomplishes two things: (1) it *reduces* heat loss, which is obvious, but from the standpoint of the comfort of the occupants it may render a far more important service in that, (2) it *increases* the inside wall or

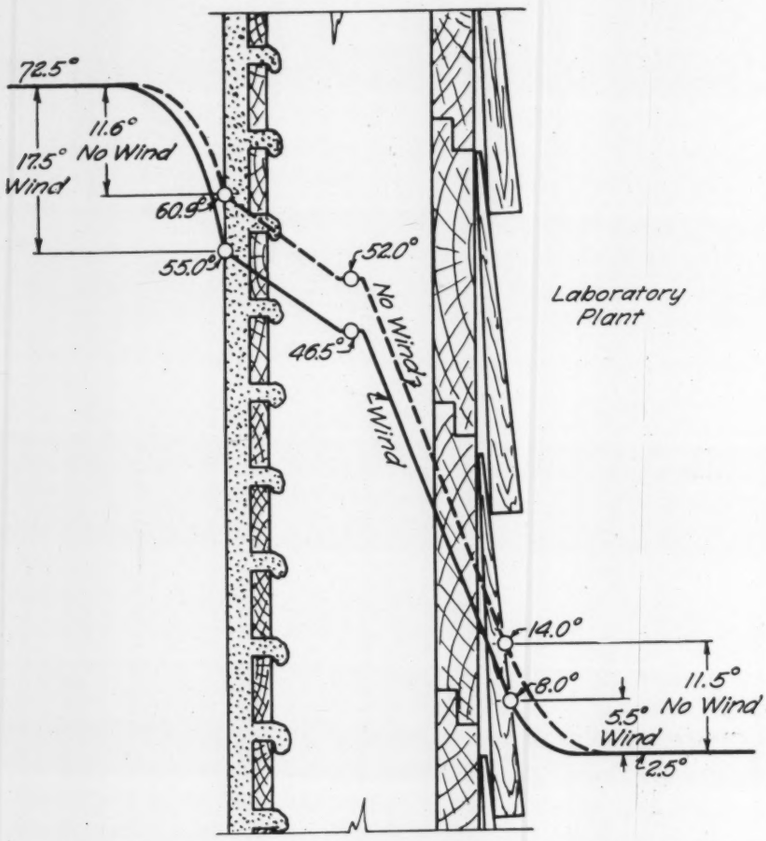


Fig. 4—Temperature Gradients Through Wall Section of Laboratory Plant

glass surface temperatures very materially in severe winter weather and thereby reduces the radiation or chilling effect of the cold wall and glass surfaces on the occupant *for the same breathing line air temperature in the room in both cases.*

(6) *The Application.*—Comparisons of the effectiveness of various types of insulation applied to building walls and glass, or of two or more uninsulated walls *after the buildings are erected and occupied and under heat can be made in the field,* by merely measuring the air and wall surface temperatures inside and outside the building. It is desirable to make these temperature measurements either at night or on a cloudy day. Under no circumstances should they be made on any wall upon which the sun is or has been shining with any intensity whatever.

By means of a portable thermocouple outfit[5] for reading surface temperatures (one inside and one outside) and the corresponding air temperatures, it is possible to check up the heat transmission and most certainly give any actual wall an exact rating in terms of some standard wall. If the measured difference between the inner wall surface temperature and the inside air temperature is less than for some standard wall, for the same inside and outside air temperatures (taken from a chart similar to Fig. 3) then it is a better wall, and a scale of values for walls and glass can easily be set up.

Moreover, the predicted performance of any wall may be checked against its actual performance as finally constructed, by comparing the actual measured surface temperatures with the calculated surface temperatures for the same wall. The calculated[6] surface temperatures are readily found for any given inside and outside air temperatures and wind movement such as may exist on the day the air and surface temperatures were actually measured. For example:

Assume two walls, No. 1 (insulated) has an overall heat transmission coefficient, $U_1 = 0.12$ B.tu per sq. ft. per hr. per degree, and No. 2 (uninsulated) has a value of $U_2 = 0.24$ B.tu. Surface coefficients will be the same in both cases, $f_{in} = 1.34$ and $f_{out} = 3 \times f_{in} = 4.02$ for the same wind movement (taken at 15 miles in both cases) and air temperatures taken at, $t = 70$ F and $t_o = 0$ F.

Wall No. 1

$$H = U_1 \times (t - t_o) = 0.12 \times 70 = 8.4 \text{ B.tu}$$
$$H = f_i \times (t - t_1) = 1.34\ (70 - t_1) = 8.4$$
$$1.34\, t_1 = 93.80 - 8.4 = 85.40$$
$$t_1 = 63.7 \text{ F inside wall surface.}$$

[5] Under *no circumstances* should the attempt be made to read these surface temperatures with ordinary mercury in glass thermometers attached to the surfaces. The authors have found that the resulting errors are relatively very large.

[6] Data for calculation taken from Chapter I of the "Guide, 1929," American Society of Heating and Ventilating Engineers.

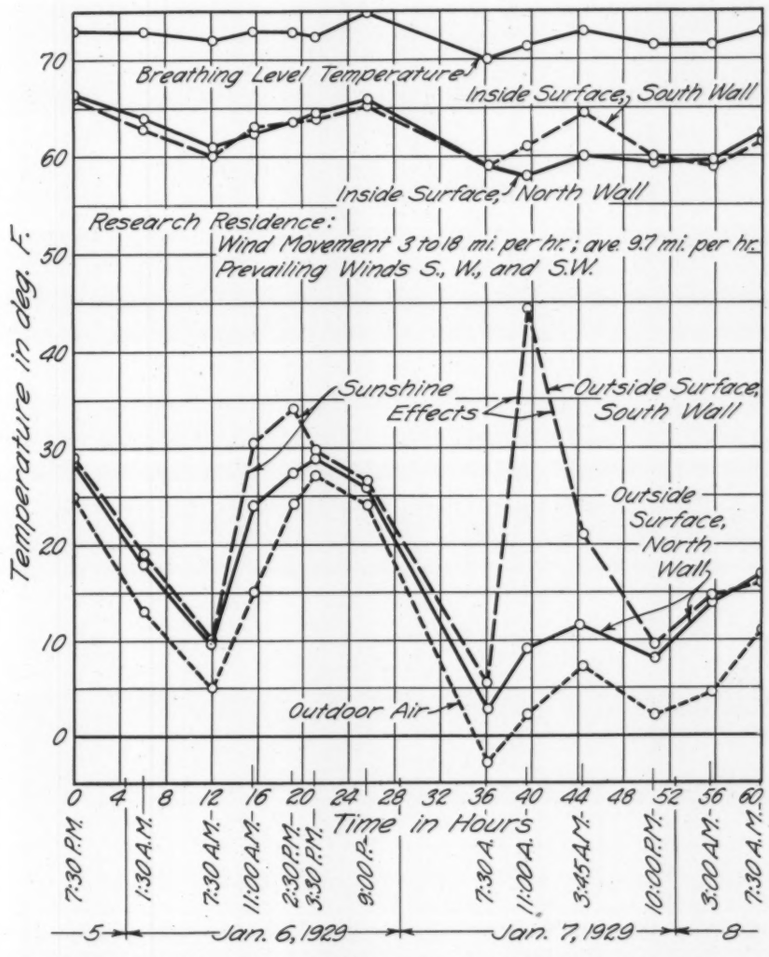


Fig. 5—Graphic Log of Air and Wall Surface Temperatures at Research Residence

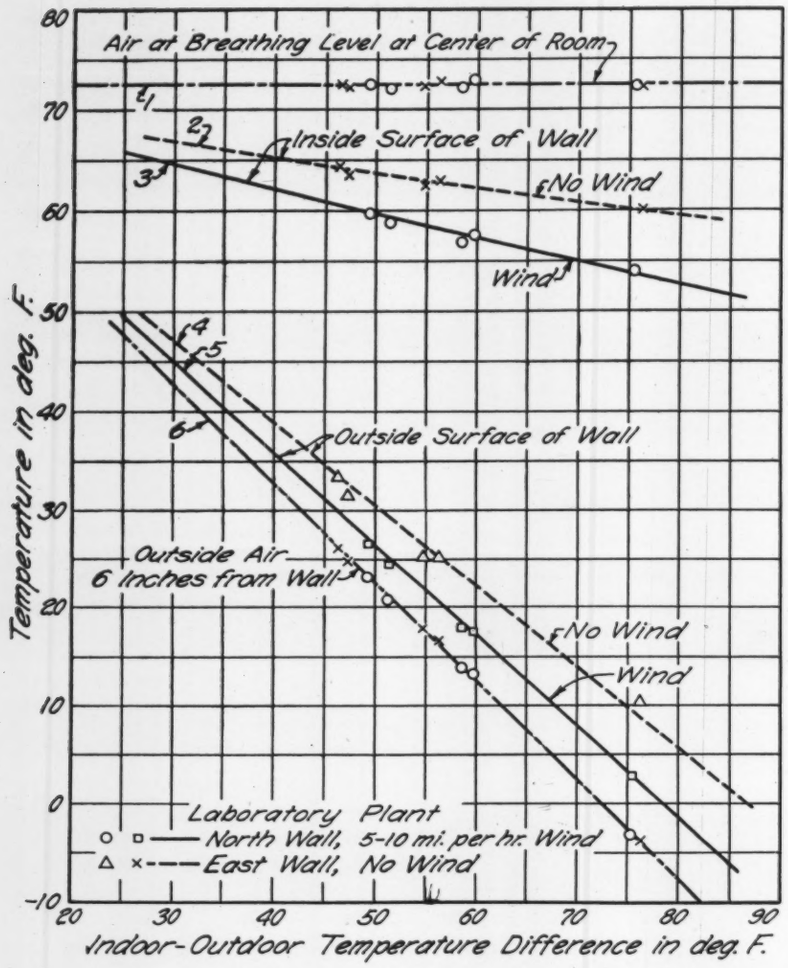


FIG. 6—INSIDE AND OUTSIDE AIR AND WALL SURFACE TEMPERATURES AT LABORATORY PLANT

$H = f_0 \times (t_2 - t_0) = 4.02\ (t_2 - 0) = 8.4$

$4.02\ t_2 = 8.4$

$t_2 = 2.09$ F outside wall surface.

Wall No. 2

$H = U_2 \times (t - t_o) = 0.24 \times 70 = 16.8$ B.tu

$H = f_1 \times (t - t_1) = 1.34\ (70 - t_1) = 16.8$

$1.34\ t_1 = 93.8 - 16.8 = 77.0$

$t_1 = 57.4$ F inside wall surface.

The outside t_2 may be calculated as before, or since

$$(t_2 - t_o) = \frac{1}{3} \times (t - t_1) = \frac{1}{3} \times 12.6 = 4.2 \text{ F.}$$

$t_2 = 4.2$ F outside wall surface.

Note:—Actual surface temperatures read directly from the curves of Fig. 5 for a 70 F indoor-outdoor difference and corrected to a breathing line air temperature of 70 F and outside air temperature of 0 F for a wall similar to No. 2 show:

t_1 (measured) = 60 — 2.5 = 57.5 F inside surface (calculated = 57.4 F).

t_2 (measured) = 9 — 2.5 = 6.5 F outside surface for an average wind velocity of about 10 miles per hour. At a wind velocity of 15 miles per hour the outside surface temperature would have been decidedly lower, and would have more nearly agreed with the calculated value of 4.2 F. It is to be noted that the inside surface temperature would have been lower also at a wind velocity of 15 miles per hour but it would only be affected about one-third as much as the outside surface temperature, and would still be in good agreement with the calculated value of 57.4 F.

It will be evident that with a portable thermocouple outfit for actually measuring t_1 and t_2 on a zero day with the breathing line temperature of the air in house at 70 F, one could readily rate the *actual* wall *as built* against walls No. 1 and No. 2. The air temperatures selected do not have to be 70 F and zero, but are whatever exists on the day of the test, although a fairly low outside temperature is desirable, and the surface temperatures should be taken on a north wall on a cloudy day or at night.

Mechanized Heating With Anthracite

The anthracite industry is assisting the development of equipment for burning the fine sizes of coal automatically, thus giving greater efficiency in the use of this type of fuel, the *United States Bureau of Mines*, Department of Commerce, points out.

While there has been remarkable progress of late years in the utilization of coal for industrial purposes, there have been until lately very few changes in the design of domestic coal furnaces. Recent advances have been made, however, in heating homes automatically with anthracite, as indicated by the number of small automatic stokers now on the market. These can burn the cheaper fine sizes of anthracite a pinch at a time.

Standard Code for Testing and Rating Steam Unit Heaters[1]

Edition of January, 1930

Adopted by The Industrial Unit Heater Association and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS[2]

IN ORDER to establish a standard method of testing and rating fan unit heaters using steam as a heating medium, this code has been prepared by a joint committee of the *Industrial Unit Heater Association* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

A. DEFINITIONS

1. **Standard Air** is air weighing 0.07488 lb. per cubic foot. (This weight corresponds to dry air at 70 F or air with 50 per cent relative humidity at a dry-bulb temperature of 68 F when the barometric pressure is 29.92 in. mercury. Specific heat is taken as 0.2415.)

2. **Static Pressure (S. P.)** is measured at right angles to the direction of air flow.

3. **The Entering Temperature** is the average temperature of the air entering the heater measured at the heater inlet and expressed in degrees Fahrenheit.

4. **Final Temperature** is the average temperature of the air discharged from the heater measured at the heater outlet and expressed in degrees Fahrenheit.

5. **Horsepower** is the brake horsepower input to the fans at a stated speed and stated conditions of rating.

6. **Steam Pressure** is gage pressure in pounds per square inch at 29.92 in. barometric pressure.

7. **Equivalent Direct Radiation (E. D. R.)** is the B.tu output at the standard basis of rating divided by 240.

B. RATING

8. **The rating** of a unit heater shall include the following:

A. R.P.M. OF FAN at full load speed

B. HEAT OUTPUT in B.tu per hour

C. AT LEAST ONE RATING showing air delivered by the heater in cubic feet per minute of standard air at standard basis of rating

D. BRAKE HORSEPOWER required by fan at standard basis of rating

E. FINAL TEMPERATURE

9. **The rating** shall state the steam pressure (at 29.92 in. barometric pressure) and the entering air temperature at which the B.tu and final temperature are taken. It shall also state the temperature at which the c.f.m. value is taken (*i. e.*, 70 F or final temperature or entering temperature).

C. STANDARD BASIS OF RATING

10. **The standard basis** of rating is to be dry saturated steam at 2-lb. gage pressure (at 29.92 in. barometric pressure) at the heater coil and air at 60 F entering the heater when the heater is free of external resistance.

11. **Each rating table** for heaters for which 2-lb. gage is a suitable working pressure, shall include one set of ratings at standard conditions and a statement of its capacity in equivalent direct radiation (E. D. R.).

D. TOLERANCE

12. **As there** are errors of measurement and inequalities of manufacture, a variation of 2½ per cent in test results will not be considered excessive.

13. **This Code prescribes** tests to determine the B.tu from measurement of condensation by weight and the c.f.m. from condensation and temperature rise, and further prescribes a means for correcting the B.tu and temperature rise, as obtained under test conditions, to any other condition of entering air temperature and steam pressure.

14. **It is required that a check** of the air quantity determined from condensation temperature rise be made by direct measurement with a calibrated nozzle. The results of the two methods shall agree within 5 per cent before the tests shall be considered correct. However, the results determined from condensation temperature rise shall govern for the purpose of rating under this code.

15. **This Code** prescribes that the heater being tested shall discharge into a receiving chamber where the air is thoroughly mixed in order that its true average temperature may be determined. The conditions within this chamber are to be controlled in such a way that the heater delivers the same capacity as when under normal conditions of free delivery.

16. **In order** to be sure that the tests on the receiving chamber are a faithful reproduction of free delivery conditions, the following routine is prescribed. First, measure the condensation of the heater during a test under actual free delivery conditions before the heater is connected with the receiving chamber. Then make three tests with the heater connected to the receiving chamber, one while holding — 0.05 in., one while holding zero and one while holding + 0.05 in. water column constant on the chamber pressure draft gage. During these tests it is necessary to measure only the condensation, entering air temperature and chamber pressure. These may each be of but ½ hour duration. These four tests shall be made at the same fan speed and at entering air temperatures within 4 deg. of each other.

17. **The condensation** measured on these three chamber tests shall be corrected to the entering temperature of the free delivery test in the proportion of the temperature differences between the steam and entering air.

18. **A curve** shall then be plotted from the three tests on the chamber showing condensation against chamber pressure. This curve shall be assumed to be a straight line within this narrow range of chamber pressures. From this curve shall be read the chamber pressure corresponding to the condensation measured on the free delivery test. The

[1] Code prepared by committee consisting of D. E. French, *Chairman*, G. E. Otis, O. K. Dyer, J. H. Shrock, W. A. Rowe, R. W. Page and L. C. Soule. This committee was assisted in its work by the A. S. H. V. E. Research Laboratory in cooperation with the Mechanical Engineering Laboratory of the University of Kentucky and by the Engineering Committee of the *Industrial Unit Heater Association*.

[2] Adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the 36th Annual Meeting, Philadelphia, Pa., January, 1930.

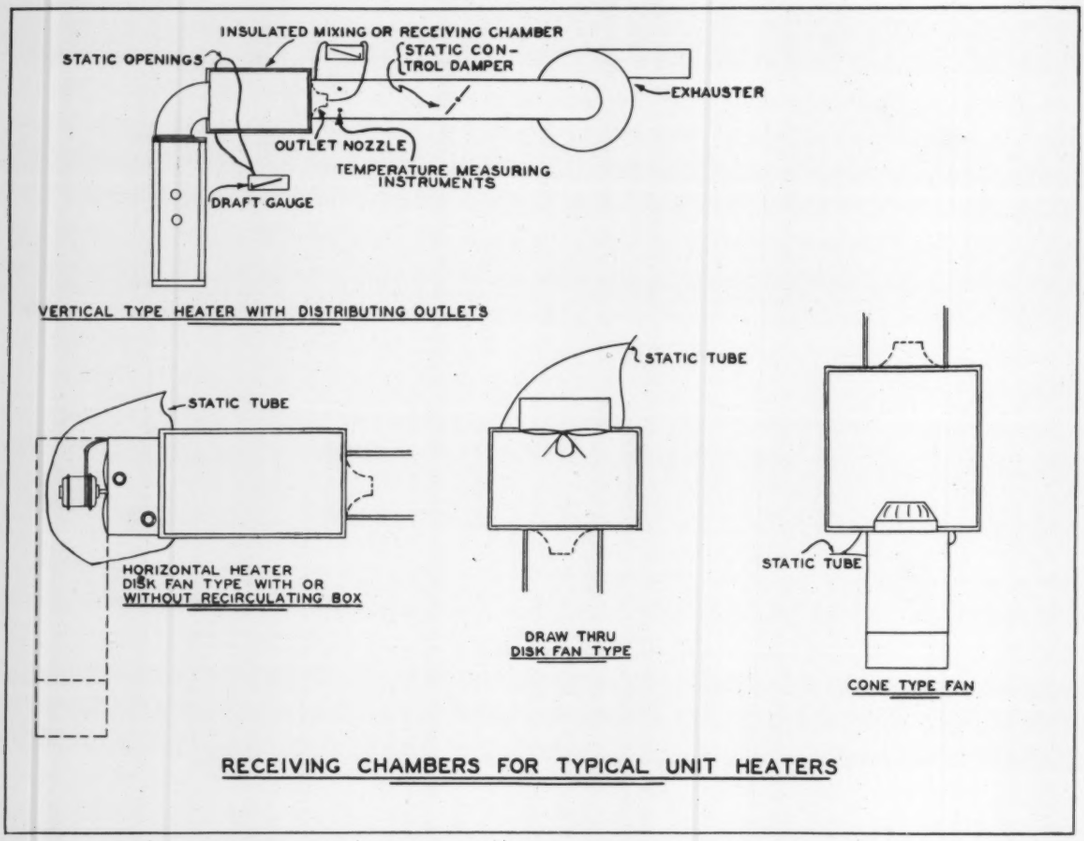


FIG. 1

draft gage which records the receiving chamber pressure, shall be held constant at the reading thus determined, during the tests necessary for rating data as described herinafter. This series of tests for the calibration of the receiving chamber to reproduce free delivery conditions shall be made for each heater at each fan speed. Should the free delivery con-

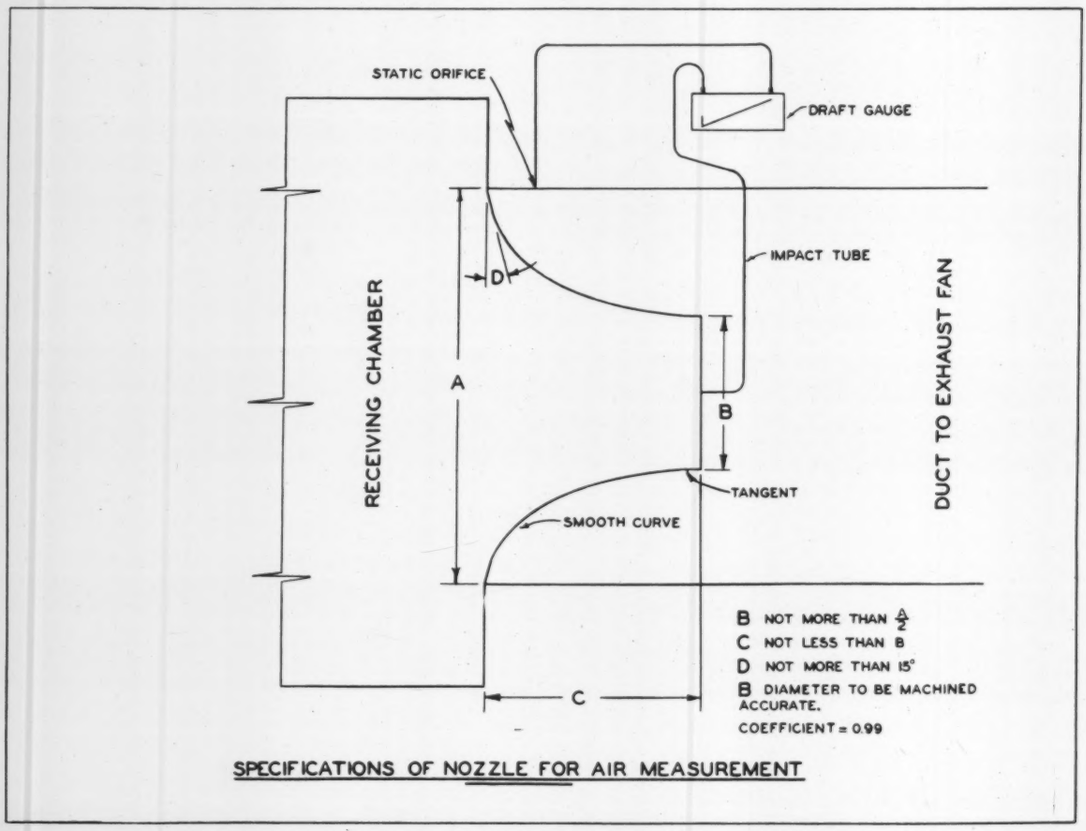


FIG. 2

densation in any series of tests meet the chamber pressure curve for that series at a point above + 0.05 or below — 0.05 it is indication that too great an error has been made either in the free delivery test or the check tests on the chamber, which must be corrected or the tests repeated.

F. EQUIPMENT FOR TESTING

19. **A chamber** for receiving and mixing the air discharged from the heater, to be constructed of any material suitable, to be made air-tight and well insulated (2-in. cork or equivalent).

20. **This receiving chamber** shall be connected by a duct to an independent exhaust fan of such capacity that it will overcome the resistance of the chamber and connections and produce a zero static at the point where the heater outlet is joined to the chamber.

21. **This receiving chamber** shall be of such size that the heater to be tested will produce from 20 to 90 air changes per minute.

22. **The outlet** or outlets of the heater to be tested shall be joined directly to the receiving chamber without a communicating duct. (See accompanying sketches of typical cases, Fig. 1.)

23. **Two or more static orifices** shall be located in the receiving chamber and not in the direct air blast from the heater. These openings shall be connected by air-tight tubes to a common draft gage which can be read to 0.005 in.

24. **Means shall be provided** with which to vary the capacity of the exhaust fan so as to maintain a zero static constant on the draft gage.

25. **A calibrated nozzle**[3] shall be fitted into one wall of the chamber, discharging into the duct leading to the exhaust fan. The outlet opening of this nozzle shall be of such area that the air velocity is not less than 3,000 f.p.m.

26. **Instruments shall be located** at the point of 3,000 f.p.m. minimum velocity for measuring the final temperature, which shall be the average of temperatures taken simultaneously at at least two points in the plane of the nozzle outlet for each square foot of outlet area, but in no case less than four points.

27. **A draft gage** shall be provided for measuring pressures at the nozzle. One side of this draft gage shall be connected to a static orifice located flush with the inner wall of the exhaust duct and near the chamber wall. The other side shall be connected to an impact tube arranged to measure the velocity pressure of the air in the nozzle outlet.

28. **The air handled** by the heater shall be disposed of in such a way as to prevent fluctuation in the temperature of the air entering the heater.

29. **Steam shall be supplied** from a source of sufficient capacity to prevent sudden changes in pressure.

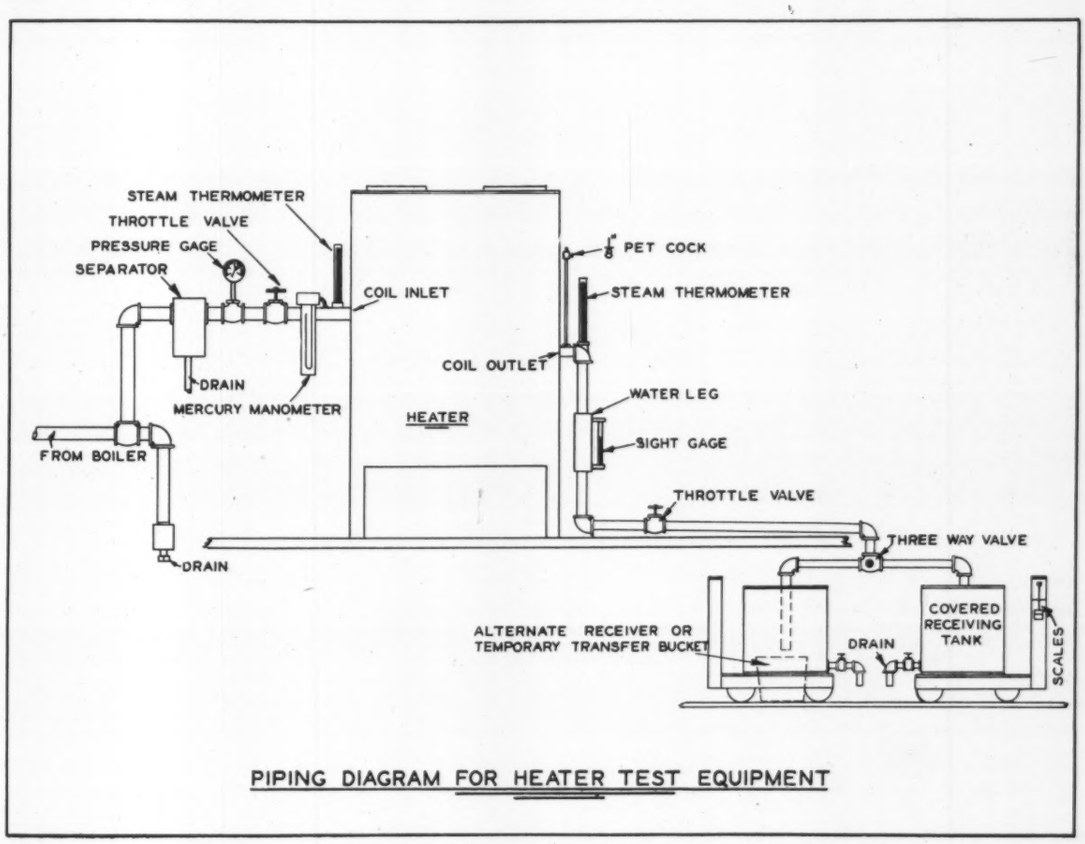
30. **The accompanying piping** diagram (Fig. 3) shows the connections and equipment prescribed for supplying steam to the coil and measuring the condensate. All of the fittings and instruments shown shall be used and shall be installed in the relative positions indicated.

31. **The separator** shall be of liberal capacity. The steam throttle valve shall be of a type suitable for close control. In selecting the type of manometer, it must be remembered that condensation will collect above the mercury on the steam pressure side and lead to error unless compensated for.

32. **The pet cock** used for air relief at the outlet of the heater coil shall be not more than ⅛ in. in size.

33. **The water leg** shall have a sight gage so the water level can be brought to the same point at the time of each reading of the condensate.

[3] If the nozzle described in Fig. 2 is used, a coefficient of 0.99 may be assumed without calibration.



PIPING DIAGRAM FOR HEATER TEST EQUIPMENT

FIG. 3

34. **The fittings and piping** from the coil outlet to the thermometer shall be insulated.

35. **The scales** for weighing condensate shall be of the beam type, capable of being read to ¼ lb.

36. **The tanks** receiving the condensate shall be covered to reduce loss by evaporation.

37. **Temperature measuring instruments** shall be disposed around the intake of the heater in such locations and in such numbers as will reflect a true average of the entering air temperature. All temperature measuring instruments shall be capable of being read to 0.5 F or closer and shall be calibrated. When exposed to radiant heat they shall be shielded therefrom.

38. **The heater casing** need not be insulated for the purpose of this code.

39. **A stop watch** shall be used for the accurate timing of readings.

40. **A barometer** shall be provided to determine the atmospheric pressure during test.

41. **A revolution counter** shall be provided to determine the r.p.m. of the fans.

G. TEST PROCEDURE

Note: To avoid duplication, this description deals with the complete rating tests as made on the receiving chamber after the control of the chamber has been checked for reproduction of free delivery conditions. However, such of this procedure as relates to the measurement of entering air temperature and condensation applies equally to the condensation tests under actual free dleivery conditions, and at alternate chamber pressures which are first made for this check.

42. **All steam and condensate lines** shall be inspected to make sure they are tight.

43. **The air-tightness of the heater connections** shall be checked, particularly of the insulated chamber up to the point at which temperature readings are taken.

44. **During the test** the steam at the heater inlet shall be held constant at a selected absolute pressure and with a superheat of not less than 2 deg.

45. **Although the test results** must later be converted to a 2-lb. basis, it has been found that a higher test pressure, for example 5 lb., provides more uniform test conditions, which is assumed in the accompanying illustrations. This applies to heaters which, when operating at a pressure of 2 lb. entering the coil, will have at the coil outlet a temperature higher than the temperature of evaporation. If the resistance to steam flow of the coil is such that steam must be supplied at a higher pressure in order to accomplish this, the heater shall both be tested and rated at this higher pressure.

46. **Turn on** the steam, open all valves and air vents wide, long enough to blow out all water. Start exhaust fan and heater fan or fans. Set control to give predetermined static in receiving chamber. Set air relief pet cock so that but a thread of steam escapes continuously. Hold steam pressure constant at the coil.

47. **An operator** should be in constant control of the throttle valve, using the manometer as his reference instrument. Before the test is begun a reference line shall be marked on the manometer corresponding to the selected test pressure corrected for the existing barometer and for the head of any water in the manometer. During the test the steam valve shall be manipulated in order to hold the mercury steady at this reference line.

48. **In order to** introduce the required superheat, there shall be not less than 5-lb. drop through the throttle valve. If the quality of steam supplied is so low that the required superheat is not introduced by the 5-lb. throttling, the line pressure shall be raised until the required superheat is introduced.

49. **Continue warming up** until the air-inlet and air-outlet conditions have stabilized to such an extent that the average temperature change is 1 F or less in 10 min. Check draft gage zero by disconnecting static tube. Reconnect and reset exhaust damper if necessary.

50. **Divert the condensate** to waste tank and determine tare on weighing tank. Be sure that scales are free in operation, and both scales and tank are free from external contacts. Mark on the sight gage of the return trap or water leg the point at which the water level is to be held, at each point of reading the condensate.

51. **Check steam pressure** and temperature, note time to the second, and divert water to weighing tank. This begins the test. The test shall be continued for one hour, during which all conditions shall be held reasonably constant.

52. **The following readings** shall be taken and recorded

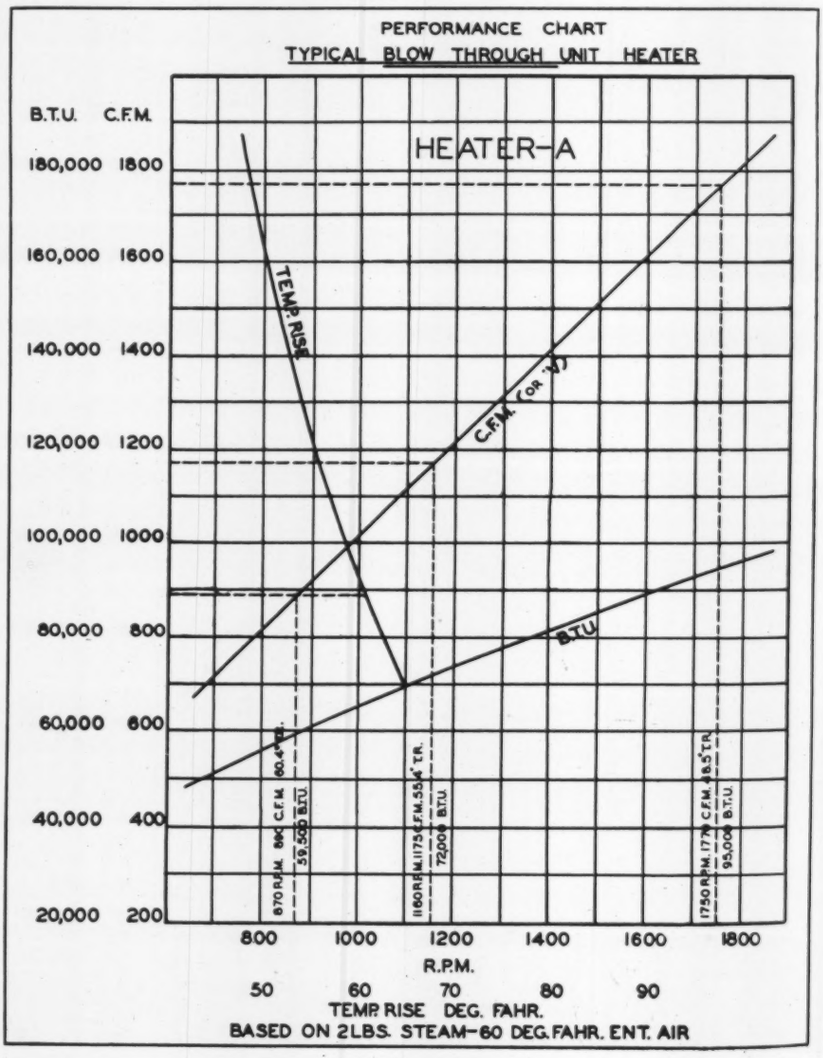


FIG. 4

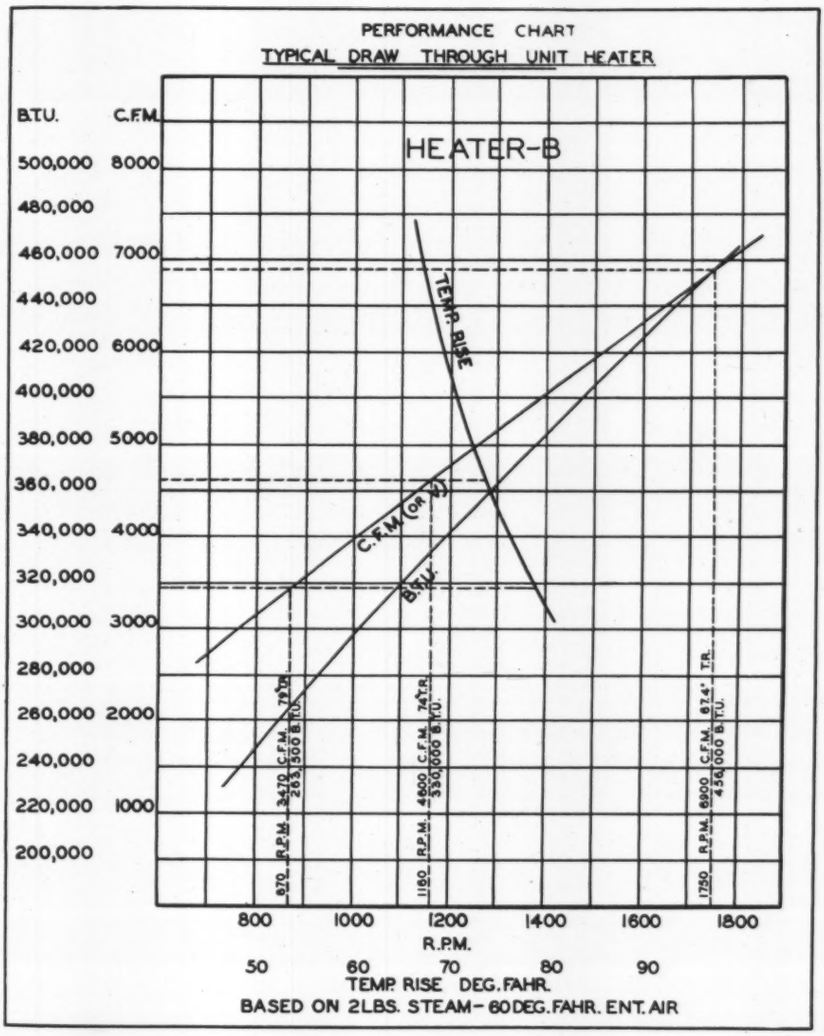


FIG. 5

at intervals of 10 min. or less:

A. Steam pressure
B. Steam temperature (coil inlet and outlet)
C. Outlet temperature
D. Inlet temperature
E. Static pressure at heater discharge
F. Weight of condensate for period
G. R.p.m. of fan
H. Hp. input to fan.
I. Differential pressure at nozzle.

53. **Exactly at** the end of each period, the condensate shall be diverted to an alternate weighing tank or change bucket. (If two tanks are used for actual weighing, they should each be tared.) The accumulation of condensate for the period shall then be weighed and recorded.

54. **If the data** recorded for successive periods are inconsistent or vary beyond a reasonable margin, the test shall be continued until one hour of consistent data are recorded. Care must be observed to bring the water level in the sight gage to the original position at each reading and to divert the condensate at the end of each reading period to the second.

55. **If the temperature** at the coil outlet falls below the temperature of evaporation for the barometer of test, it indicates either that air is not being removed from the coil or that the resistance of the coil is too high for the selected test pressure.

56. **The horsepower input** to the fans may be computed from the watts input to the fan motor if a calibrated motor is used.

H. COMPUTATION OF RESULTS

57. **Chamber calibration tests.** The condensation obtained on each of the receiving chamber check tests C shall be converted by the following formula to its approximate equivalent C', at the entering air temperature t'_e of the test made at the same fan speed under free delivery conditions:

$$C' = C \frac{t'_s - t'_e}{t_s - t_e}$$

where t_s and t'_s are the steam temperatures of the receiving chamber test and the free delivery test, respectively, and t_e is the entering air temperature of the receiving chamber test.

58. **Check test of c.f.m.** The volume of air, in cubic feet per minute, passing through the nozzle, at final temperature as of test conditions, shall be calculated by the following formula:

$$\text{Volume} = 1096A \sqrt{\frac{VP}{w}} \times K$$

where A is the outlet area of the nozzle in sq. ft., VP is the differential pressure across the nozzle, w is the weight of air at the temperature in the nozzle and the barometer of test, and K is the coefficient of the nozzle.

59. **Rating tests.** The following result directly from test data:

A. Average temperature of entering air in degrees Fahrenheit, t_e
B. Average temperature of leaving air in degrees Fahrenheit, t_f
C. Temperature rise of air, $t_r = t_f - t_e$
D. Saturated temperature of steam in degrees Fahrenheit, t_s
E. Pressure of steam, pounds per square inch, gage, p
F. Barometric pressure, pounds per square inch, b
G. Total or absolute pressure of steam, $P = p + b$
H. Pounds of condensation per hour, C
I. Latent heat of steam, R

60. **The following** are calculated directly from test data:

A. B.tu per hour as of test, $H = CR$ (1)

B. C.f.m., standard air, as of test, $M = \frac{55.3\,R\,C}{60t_r}$ (2)

61. **It is generally** impracticable to test heaters under exact predetermined entering air temperatures and steam pressures. Since, however, it is necessary to rate them under stated conditions, a standard procedure is given whereby data of test may be used for the determination of performance under any such desired conditions of rating. For this purpose certain basic assumptions are made.

62. **It is assumed** that the volume of air handled by the fan or fans at the temperature in the fans, is constant for a given heater and fan speed regardless of temperature changes.

63. **The following graphic method** of solution is prescribed. It requires at least three separate tests on each heater, each at a different fan speed, in order to secure enough points to establish the lines. The speeds selected—

one low, one medium and one high—shall embrace the entire range of speeds for which the heater is to be rated.

A. First plot a line, designated as the c.f.m. or V line, as illustrated in Figs. 4 and 5, showing the volume of air in cubic feet per minute V handled by the fan or fans, at the temperature in them, at various speeds.

B. Points for establishing this line shall be computed from test data by the following formulae:

$$\textit{For Blow-Through Heaters, } V = \frac{(t_e + 460)}{(70 + 460)} M \qquad (3a)$$

$$\textit{For Draw-Through Heaters, } V = \frac{(t_f + 460)}{(70 + 460)} M \qquad (3b)$$

C. Having determined this line, find by means of it at what speeds the fan or fans would have to run to handle an equal weight of air, in each case, at prescribed entering air temperature t'_e and steam temperature t'_s. The equivalent volume V' of this equal weight of air under prescribed conditions shall be computed as follows:

$$\textit{For Blow-Through Heaters, } V' = \frac{(t'_e + 460)}{(70 + 460)} M \qquad (4a)$$

$$\textit{For Draw-Through Heaters, } V' = \frac{(t'_f + 460)}{(70 + 460)} M \qquad (4b)$$

D. In solving Formula 4*b* it is necessary to determine the final temperature t'_f. Since for the same weight of air the temperature rise is proportional to the differences between the steam and entering air temperatures, the temperature rise $t'r$ for the desired entering air and steam temperature condition is as follows:

$$t'_r = t_r \frac{(t'_s - t'_e)}{(t_s - t_e)} \text{ and} \qquad (5)$$

$$t'_f = t'_r + t'_e$$

E. Next determine the B.tu per hour that would have been developed in each case of test had the same weight of air been handled at other prescribed entering air and steam temperatures, as follows:

$$\text{B.tu} = \frac{60\, M t'_r}{55.3} \qquad (6)$$

F. Now determine the B.tu line by plotting these B.tu values against the speeds (r.p.m.) at which the heater would have to operate, in each corresponding case, to handle an equal weight of air, as of test, at the prescribed entering air and steam temperatures.

G. In order to take off complete rating data at any specified heater speed it is desirable to add a line showing temperature rise, under prescribed entering air and steam temperature conditions, plotted against values of V.

H. For determining the points that establish this line, the temperature rise in each case shall be found by means of Formula 5. These values shall be plotted against corresponding V' values obtained by Formula 4*a* or 4*b*.

I. The capacity of the heater in B.tu per hour and the corresponding capacity in c.f.m. of air measured at the temperature in the fan or fans both at prescribed entering air temperature and steam pressure and at any desired fan speed, can then be read directly from the chart.

J. To do this, trace vertically upward along the line representing a selected speed to the points of intersection with the V and B.tu lines. Then trace horizontally from these points of intersection to find c.f.m. at temperature of air in fans (c.f.m.$_{t_{an}}$) and B.tu per hour on their respective scales.

K. To find temperature rise, trace horizontally from the intersection of the speed line and V curve to point of intersection with temperature rise curve. Then drop vertically down to temperature rise scale. This temperature rise plus the prescribed temperatures gives the final air temperature.

L. The consistency of the values taken in this manner from the chart should be checked by equating them in the following formulae:

$$\textit{Blow-Through, } \text{B.tu} = \frac{60\,(460 + 70)\ \text{c.f.m.}_{t_{an}}\, t'_r}{55.3\,(460 + t'_e)}$$

$$\textit{Draw-Through, } \text{B.tu} = \frac{60\,(460 + 70)\ \text{c.f.m.}_{t_{an}}\, t'_r}{55.3\,(460 + t'_f)}$$

M. If they do not check closely for each selected speed, the computations and plotting and, if necessary, the tests themselves should be carefully looked over for errors. If the discrepancy is negligible, however, the values may be brought into agreement by arbitrary adjustment of the temperature rise.

N. If desired, the c.f.m. at temperature of air in the fan may be converted to c.f.m. of standard air as follows:

Blow-Through

$$\text{c.f.m.}_{std.} = \text{c.f.m.}_{t_{an}} \frac{(460 + 70)}{(460 + t'_e)}$$

Draw-Through

$$\text{c.f.m.}_{std.} = \text{c.f.m.}_{t_{an}} \frac{(460 + 70)}{(460 + t'_f)}$$

SAMPLE DATA—RECEIVING CHAMBER TEST

Heater A, Blow-Through Type

Speed of fan at test	1772	1177	859
Entering Air Temperature at Test (t_e)	73.4	65.5	75.0
Final Temperature at Test (t_f)	120.3	122.0	132.7
Temperature Rise as of Test (t_r)	46.9	56.5	57.7
Condensation per hour at test (C)	94.5	75.75	59.5
Saturated steam temperature at coil (t_s)	230	230	230
Absolute Pressure (P)	19.7	19.7	19.7
B.tu as of test $C \times R$	90,700	72,750	57,200
C.f.m. @ 70 as of test (M) $M = \frac{(55.3 \times R \times C)}{(60 \times t_r)}$ =	1785	1188	914
C.f.m. @ fan $V = M \frac{(460 + t_e)}{(460 + 70)}$ =	1798	1178	922
Temperature Rise 60 F Entering 2 lb. steam Same weight of air as of test $t'_r = t_r \frac{(t'_s - t'_e)}{(t_s - t_e)}$ =	48.6	55.7	60.4
Volume at fan with 60 F Entering Same weight of air as of test $V' = M \frac{(460 + t'_e)}{(460 + 70)}$ =	1753	1165	897
B.tu with 60 F Entering 2 lb. steam Same weight of air as of test B.tu $= M \frac{(t'_r \times 60)}{(55.3)}$ =	94,100	71,800	59,900

Heater B, Draw-Through Type

Speed of fans at test	1743	1179	899
Entering Air Temperature at Test (t_e)	86.3	85.8	90.3
Final Temperature—Fan Temperature (t_f)	146.3	151.7	158.4
Temperature Rise as of Test (t_r)	60.0	65.9	68.1
Condensation lb. per hour test (C)	407.74	362.0	235.75
Saturated steam temperature at coil (t_s)	229.5	229.5	229.5
Absolute Pressure (P)	19.7	19.7	19.7
B.tu as of test—$C \times R$	391,500	290,000	226,000
C.f.m. @ 70 as of test $M = \frac{(55.3 \times R \times C)}{(60 \times t_r)}$ =	6030	4060	3060
C.f.m. @ fans $V = M \frac{(460 + t_f)}{(460 + 70)}$ =	6900	4690	3570

Temperature Rise @ 60 F Entering 2 lb. steam

Same weight of air as of test

$$t'_r = t_r \frac{(t'_s - t'_e)}{(t_s - t_e)} = 67.9 \quad 74.2 \quad 79.2$$

Volume at fans with 60 F Entering 2 lb. steam

Same weight of air as of test

$$V' = M \frac{(460 + t'_f)}{(460 + 70)} = 6690 \quad 4560 \quad 3460$$

B.tu with 60 F Entering same weight of air as of test

$$\text{B.tu} = M \frac{(t'_r \times 60)}{(55.3)} = 444{,}500 \quad 327{,}000 \quad 263{,}500$$

Note: In writing this code, the committee had in mind the need for commercial simplicity and practicability, eliminating from the test procedure and computations certain complications which, while they would promote greater accuracy, are not, however, essential in producing results that are accurate within the tolerances established for this code. These are, however, discussed in the addenda.

ADDENDA

STANDARD CODE FOR TESTING AND RATING STEAM UNIT HEATERS

A1. These addenda include

A—Recommendations for the construction and use of the equipment prescribed by the Code, discussed in more detail than was feasible in the Code itself.

B—Discussion of refinements by which the Code test procedure can be elaborated by those interested in a higher degree of accuracy than the code requires.

Recommendations—Code Equipment and Its Use

A2. A receiving chamber is prescribed by the Code in order to effect a thorough mixing of the air discharged from the heater so that its true average temperature can be read accurately, but the conditions in this chamber shall be controlled in such a way that the normal free delivery conditions of heater operation are duplicated.

A3. Even though insulated, this chamber shall be relatively small, so the heat loss through its walls, which is neglected by the Code, will be a minimum. This leads to the specification in the Code of 20 to 90 air changes per minute. Yet the chamber must be large enough so that the velocity of the discharged air is reduced to 40 per cent (or less) of the outlet velocity through the net outlet area of the heater. This is desirable for accurate control of the pressure in the receiving chamber to produce free delivery conditions. It is further desirable that the discharge air pass from the heater outlet to the point where its temperature is measured in less than three seconds.

A4. With these requirements in mind the size and proportions of a suitable mixing chamber can readily be designed for a given size of heater. However, the manufacturer who has a number of sizes to be tested may want to provide a chamber, the proportions of which can be readily adjusted rather than provide two or more chambers for his range of heater sizes. For this purpose it is recommended that the chamber be designed in the proportions that are correct for the largest heater, but that it be provided with two insulated partitions to be placed parallel with the air flow so that the width of a central passage through the chamber can be varied to suit the requirements of the smaller heaters.

A5. Slides to receive these partitions can be built into the chamber so that the partitions can be moved from one to the other readily as required. An adapter plate will be required for each size heater to connect that heater to the fixed opening of the chamber. The openings in this adapter plate to receive the heater outlets should be so located that the heater outlet or outlets are centered in the receiving chamber opening.

A6. When the heater is operating under normal conditions of free delivery, there is a zero static pressure around the periphery of the heater outlet. This is the condition to be reproduced within the receiving chamber. In order to give a true reflection of this condition, the static orifices should be located in or near the plane of the heater outlet, and within 6 in. of the periphery of the heater outlet and not in the path of the air, *i. e.*, not at a position where the static orifice can register velocity pressure.

A7. Orifice plugs set into the walls of the chamber with their faces flush with the interior surface of the walls have been found to give good results. For housed fan heaters or disc fan blow-through heaters, these orifices should be located in those walls of the chamber that are parallel to the flow of the air as it passes through the chamber. For disc or cone type fan draw-through heaters, where the fan itself is inside the chamber and has a tendency to throw off air radially, the orifices should be located in the end wall which is in or back of the plane of the heater outlet. No more exact requirements for the positioning of these orifices can be given in view of the wide range of the types of heaters in common use.

A8. While it is correct to assume that free delivery conditions are reproduced if zero static is maintained around the periphery of the outlets when the heater is connected to the chamber, it must be remembered that unlooked-for eddy currents within the chamber might influence the static orifices, no matter how well their location is selected, so that they would give a false reading of the true static condition. To guard against error from this source, a series of check tests is provided in the Code, so that the rating tests can be run at that reading of the static orifice draft gage at which the free delivery condensation of the heater is found to be duplicated. This reading might not be zero, but it should approach it. The fact that it may not be zero is accounted for by the possible influence of eddy currents or by the presence of unavoidable error in the check tests from which this reading was determined.

A9. If the reading indicated by the check tests as reproductive of free delivery conditions exceeds + or —0.05 in. water column, it indicates either that errors in the check tests are greater than allowable or that some unexpected condition within the chamber has an undue influence on the static orifices. If this occurs a repetition of the tests usually shows that the former is the cause. It may be necessary, however, to re-position the static orifices.

A10. Since each size of heater at each fan speed produces a different condition within the chamber, it is correct to repeat the check tests for each fan speed of each heater and the Code so provides when rating tests are to be made.

A11. Once the position of the static orifices is approved through check tests on one size of heater, it will be found that pressure readings indicated by check tests on other sizes of heaters will fall within a narrow range above and below zero. If so, check tests may be omitted from routine tests (not rating tests), without objectionable error, holding zero static uniformly throughout.

A12. The Code requires a check measurement of air quantity by means of a nozzle. Any form of nozzle or orifice may be used if it is properly calibrated, but the form described in the Code is recommended and, if made to Code specifications, need not be calibrated.

A13. While 3,000 f.p.m. is the minimum for the velocity of the air at the nozzle outlet, there is no maximum limit to this velocity except the practical one of the pressure required of the exhaust fan to overcome the high resistance of the nozzle at extreme velocities. If the exhaust fan has sufficient capacity to handle the air from the largest heater against the resistance of chamber and duct plus a resistance of approximately 2¼ in. for the nozzle, only two nozzles need be provided for each chamber, one of one size for the first part of the range of the chamber and one of another size for the second part of the range. If desired to use an

exhaust fan of lower pressure, one or more additional sizes of nozzle can be provided.

A14. It is important to prevent fluctuation in the temperature of the air entering the heater during the test and to prevent undue stratification of the air entering a heater having its inlet in a vertical plane. The size and arrangement of test rooms varies so widely that no specific recommendations are possible, but in general it is well to arrange the discharge from the exhaust fan so that most of the heated air is discharged out of doors or into a room that does not communicate with the room from which the heater draws its air. It is well also to admit outside air to the heater through openings so large that their resistance is negligible, as outside temperatures vary but slowly and stratification will be a minimum.

A15. In selecting the type of manometer, it must be remembered that condensation will collect above the mercury on the steam pressure side and lead to error unless compensated for. It is recommended that the manometer be so constructed that a head of water is maintained above the mercury on the steam-pressure side, so arranged that its level will be maintained constant by means of an overflow return to the steam manifold. This water column should be confined in a chamber having a liberal cross sectional area by comparison with that of the manometer tube, so that changes in the level of the mercury column will affect the water level but little. The overflow level should be an accurately measurable distance from the scale of the manometer for accurate calibration.

A16. Where Thermocouples are used as temperature measuring instruments, the cold junction should be immersed in a medium kept at the temperature of melting ice, in order to increase the differential between the hot and cold junctions and thus provide for greater accuracy.

A17. Where heaters of larger capacity are being tested, the condensate in the weighing tanks accumulates with some rapidity and it is therefore advisable to provide weighing tanks with a rapid emptying device so that the condensate of one receiver can be disposed of before the other receiver is filled.

A18. For the check test, in order to produce free delivery conditions conveniently and, at the same time, to remove from the test room the heated air discharged from the heater, it is suggested that the heater be set up with the heater outlets on a level with and directed toward the receiving chamber opening but separated from it by a distance of not less than 3 ft. Thus the air delivery of the heater will not be influenced either by the proximity of the receiving chamber or the operation of the exhaust fan which should be run at a capacity somewhat greater than the air capacity of the heater. Also the heater can then be moved into position for attachment to the receiving chamber with a minimum of work, particularly if the piping connections are arranged with this in mind.

Provisions and Corrections For Greater Accuracy

A19. Some of the water condensed by the unit during test escapes through evaporation from the receiver. This loss is neglected in the Code after being minimized by covering the receiver. It may further be reduced by providing a cooling device in the return line to the receiver. Any error through loss by evaporation tends to give B.tu and c.f.m. ratings which are less than actual.

A20. The Code does not require a correction for humidity. If the humidity at the time of the test is lower than standard and no correction is made, the B.tu rating of the heater will be less than its actual capacity when the entering air is standard humidity and *vice versa.*

A21. The Code prescribes that the air relief valve shall be slightly cracked to allow a thread of steam to escape in order to make sure that the coil is kept free of air. It would be possible, if any appreciable amount of steam escaped in this way, for this steam to carry with it a slight amount of water that would otherwise increase the weight of condensate chargeable to the heater. If a small pet cock is installed so that but a thread of steam escapes, it is considered that error from this source is negligible. Any such error is in the direction of low B.tu and c.f.m. ratings. Any such error might be avoided by substituting a thermostatic air valve for the pet cock, but it is then necessary to watch more closely for evidence of air binding.

A22. The Code neglects the heat added to the air by the power expended to move the air through the heater. This makes for low B.tu and c.f.m. ratings. For the purpose of the Code, this error is neglected because it is so small in magnitude and because it partially or completely offsets loss by radiation through the heater casing, which is opposite in sign in its effect on the air quantity rating, and which is also neglected in the Code.

A23. For extreme accuracy the heater casing should be insulated and a further correction made for the heat loss through this insulation. Any unaccounted for loss from this source results in a higher indicated air quantity rating than actually delivered. If correction is made for this loss a correction should also be made for power expended to move the air.

A24. In some heaters there may be a cooling of the condensate in the coil appreciably below the temperature corresponding to the pressure at the coil inlet. This transfer of heat from the liquid is not credited to the heater in the Code procedure which credits only the latent heat. Also the Code method neglects any heat of superheat.

A25. If greater accuracy is required, the temperature of the condensation may be taken and the heater credited with the difference in total heat of the steam entering the coil and the water leaving the coil. It is sometimes difficult, however, to maintain such constant conditions in the coil as will provide a constant temperature of the condensate, unless the steam pressure used for the test is materially above that suggested in the Code for low pressure heaters.

A26. Any error from this source produces a rating that is less than the actual, but that in amount is negligible for the purpose of the Code.

A27. The Code requires no correction for the barometric pressure or humidity in applying the factor 55.3. This error would influence the air capacity rating in either direction, depending on whether the barometer and humidity during test were high or low. For greater accuracy, use the factor correctly computed for the humidity and air density as of test.

A28. In addition to assuming a constant volume of air at the fans for a given fan speed, regardless of changes in the temperature of the air, the Code further assumes that the fan motor speed will not change when handling a given quantity of air at the varying densities of the air that will result from varying its temperature.

A29. This will not be true in practical operation where heaters are equipped with motors whose speeds vary with the load. This fact has no bearing on the Code test procedure and the methods of computations which deal only with fan speeds, without consideration of the size and characteristics of the motor necessary to produce those speeds and which result in true ratings for a given entering air temperature, steam pressure and fan speed. For greater accuracy in rating, however, it would be necessary to determine the speed at which a given motor would drive the fans when handling air at the temperature which would result with a given condition and rate the heater at that speed.

Officers and Council Members

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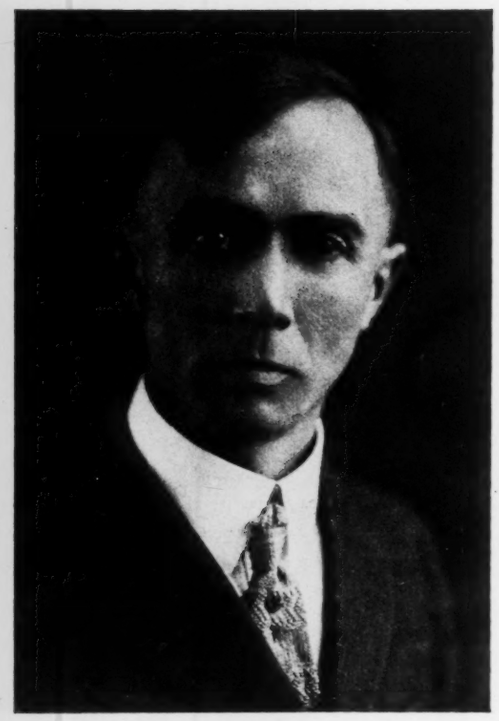
President

W. H. CARRIER
Newark, N. J.
First Vice President



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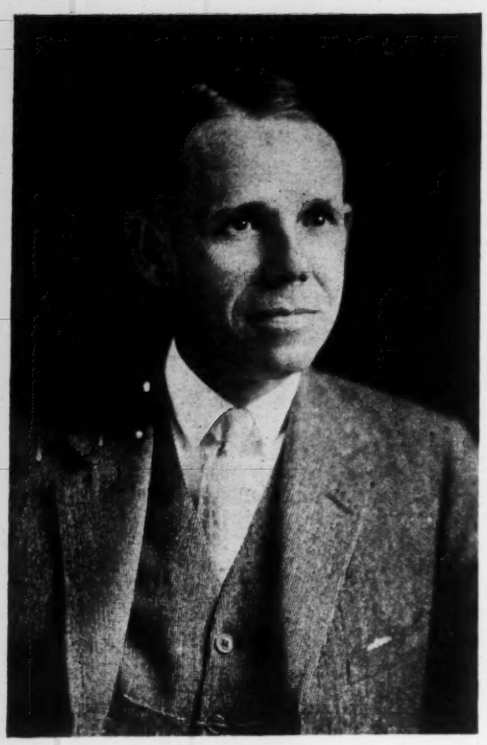

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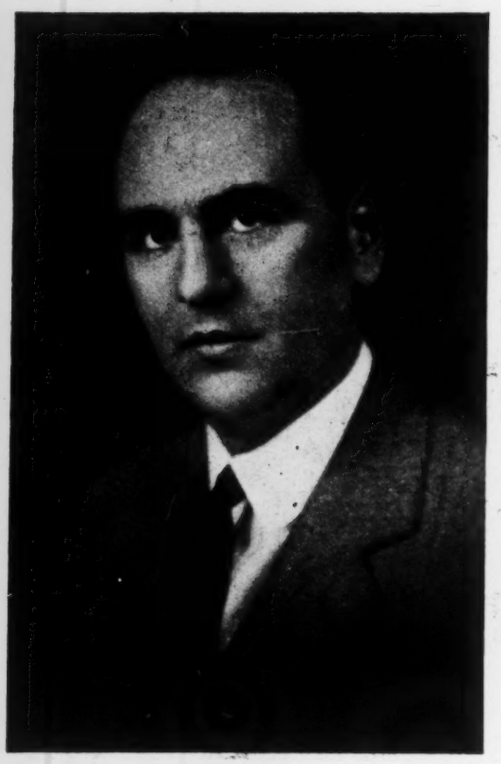

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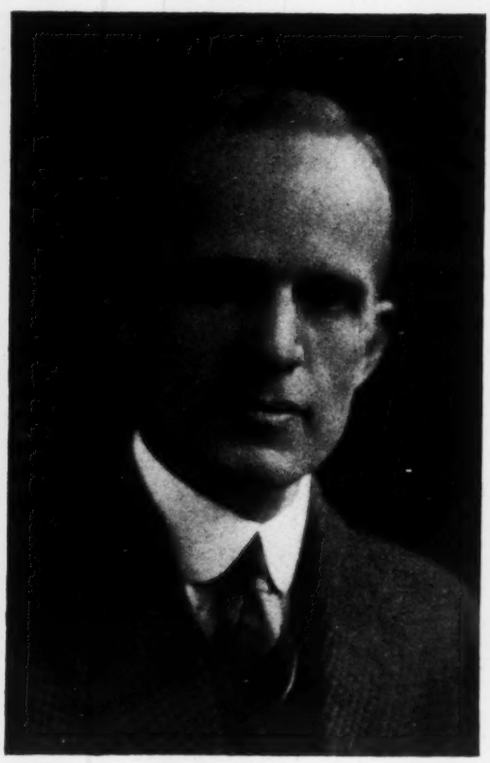
Pittsburgh, Pa.



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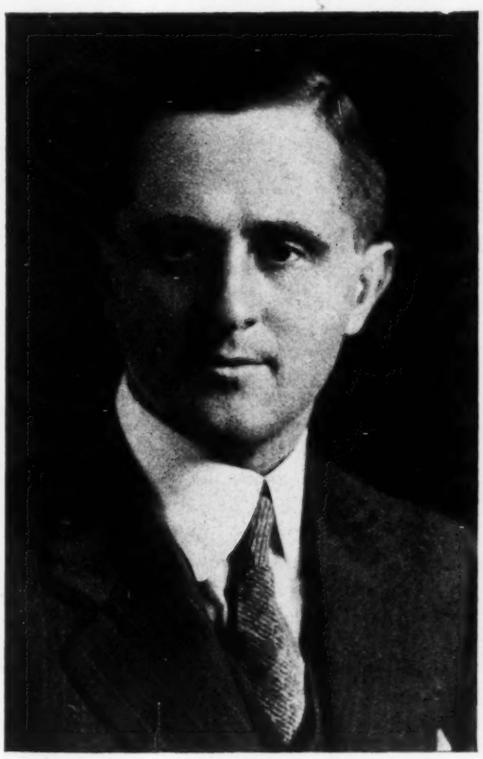
N. W. DOWNES
Kansas City, Mo.

Members of the Council for One Year



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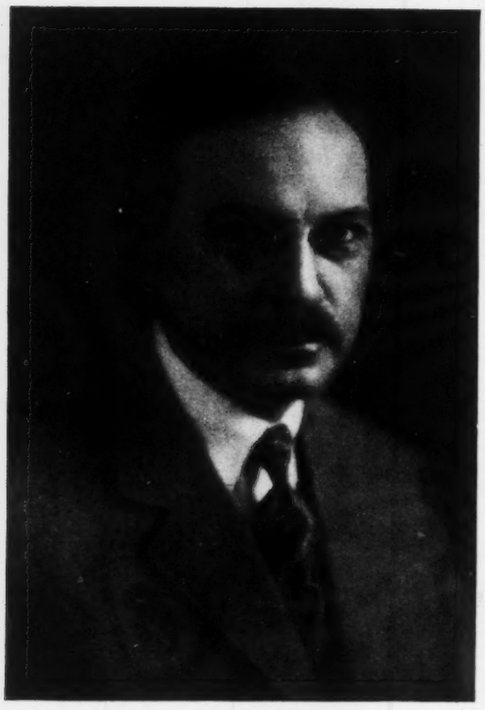

THORNTON LEWIS
Philadelphia, Pa.
Chairman of Advisory Council

Local Chapter Reports

Cleveland

February 11, 1930. The meeting was called to order by Pres. W. C. Kammerer with an attendance of 65 members and guests.

The reading of the minutes of the previous meeting and all business was suspended, except a discussion of the proposed meeting of the SOCIETY in Cleveland in 1932.

A motion by J. J. Mason, seconded H. M. Nobis, carried requesting that the Council hold the 1932 meeting in Cleveland if the Exposition could be held at the same time.

The meeting was held in the Cleveland Bldg. of the Johnson Service Co., and an interesting talk was given on temperature and humidity control by M. F. Rather, who illustrated his talk with lantern slides.

C. F. Eveleth made a motion that the Chapter express its appreciation of Mr. Rather's talk by a rising vote of thanks.

After the talk there was a bridge contest.

Michigan

February 17, 1930. The Detroit Edison Co. entertained the Chapter at their Delray Power Plant. One hundred and sixty members and guests were in attendance.

E. E. Dubry, of the Detroit Edison Co., welcomed the Assembly and then introduced Axel Hellstrom of Chicago, who gave an exhibition in mind reading and a memory test which was nothing short of miraculous. Following this, B. W. Thompson of the Detroit Edison Co. discussed many mechanical features connected with the Delray Power Plant. Some of the high lights in Mr. Thompson's talk were as follows:

The operating steam pressure is 400 lb. with a steam temperature of 725 F. The steam piping is of 600-lb. seamless steel construction and the fittings are of cast steel. Pipe expansions as high as 6 in. in 100 ft. have been noted in this plant. No high pressure steam lines are welded but some of the low pressure service lines are welded. Each stoker consumes an average of 17 tons of coal per hour. Mr. Thompson went into details as to the cost of transmission of power (about 55 per cent of the total cost of a kw) and called attention to the fact that the Delray turbines are interchangeable with their Trenton Channel Plant.

Sabin Crocker, also of the Detroit Edison Co., next delivered an address on heating and ventilating the Delray Power House. In general, air is pulled into the turbine room through windows in the summer and exhausted by means of a fan exhaust at the roof line, he stated. The exhaust fan is so connected that this air can be discharged to the forced draft fans in the winter time or can be exhausted to the atmosphere in warm weather. In cold weather the new air, instead of being drawn through windows, is drawn through heater coils so that the air is tempered before it enters the building.

The forced draft fans in the power house also draw hot air from the boiler room which, of course, tends to keep the boiler room from being so intensely hot and at the same time allows the forced draft fans to discharge tempered air instead of cold air to the boilers.

Mr. Crocker has developed an idea which is meeting with much success and that is the window sweeping system which is employed in their turbine room which eliminates, to a large degree, the formation of condensation on the ceiling of the turbine room and on the window glass. Air is drawn from the outside by means of a full housed type of fan through heater coils, and carried through ducts which are built into the building construction and then to a slot in the window ledge and discharged at a high velocity over the window. There is a variable control so that the air quantity can be adjusted.

After Mr. Crocker's address, William Boales thanked the Detroit Edison Company for their hospitality and then the entire assembly was taken through the Delray Power Plant on a trip of inspection where observations were made of the different features which had been brought out in the addresses by Messrs. Thompson and Crocker.

The Chapter was appreciative of the opportunity of inspecting the Delray Power Plant and is indeed grateful to its host, the Detroit Edison Company.

Western New York

February 15, 1930.—The February meeting of the Western New York Chapter took the form of a dinner-dance held at the Hotel Buffalo, Buffalo, N. Y., on Saturday evening, February 15.

About 35 couples were in attendance and all reported an enjoyable evening.

The officers for the new year were installed with appropriate ceremony.

President Burke spoke briefly and asked the support of all the members to his administration, which he pledged would be one of industry and activity.

The following committees were announced by Mr. Burke: *Speakers Committee,* O. K. Dyer, Walter Johnson, L. A. Cundall; *Entertainment Committee,* J. J. Landers, *chairman,* L. N. Castin, H. B. Roarke; *Membership Committee,* Joseph Davis, *chairman,* A. K. Tinker, Wm. Roebuck, Jr.

It was decided to hold meetings this year on the second Monday of the month, and at the Hotel Buffalo, until further notice is given.

Philadelphia

February 13, 1930. The February meeting was held in the faculty dining room of the Gratz Senior High School.

J. D. Cassell, who was the principal speaker of the evening, gave a brief description of the mechanical equipment installed in the Gratz Senior High School, and explained that the power plant of this building provides heat not only for the Gratz Senior High, but also for the Gillespie Junior High which is immediately adjacent, and that a total of about 6,000 students are enrolled in the two schools. He then gave a comprehensive review of the history of the various types of heating and ventilating systems installed in Philadelphia Schools during the past forty years, contrasting the old hot air system with the modern up-to-date type in which clean, washed air, in abundant quantities, is delivered to each class room, office, and auditorium in which either students or faculty meet.

At the conclusion of Mr. Cassell's talk, an inspection of the Gratz Senior High was conducted by the members of the Board of Education Engineering Department.

The following Committee Chairmen, appointed by President Black, were announced: Meetings Committee, M. F. Blankin, *Chairman;* Membership Committee, A. J. Nesbitt, *Chairman;* Committee on Attendance, W. P. Culbert, *Chairman;* Finance Committee, Lee Nusbaum, *Chairman;* Year Book Committee, C. D. Graham, *Chairman;* Legislative Committee to serve for three years, J. D. Cassell, *Chairman.*

Upon motion by Mr. Sanbern, a rising vote of thanks was extended to R. C. Bolsinger and his committee for the very efficient and competent manner in which all arrangements and details, in connection with the annual meeting of the SOCIETY, were handled.

Mr. Blankin, Chairman of the Meetings Committee, announced that the regular March meeting of the Chapter will be held at the Engineers Club, Thursday, March 13th, and that a paper covering Gas Heating will be presented.

Wisconsin

February 17, 1930. The regular monthly meeting of the Wisconsin Chapter was held Monday, February 17, at the Milwaukee Elk's Club.

President Weimer opened the meeting by calling upon John S. Jung to report on the recent Annual Meeting held in Philadelphia.

Howard Haupt, who was in charge of the February meeting, secured Homer R. Linn, American Radiator Co., as the speaker of the evening. Mr. Linn spoke on The Status of the Oil Burner in the Heating Industry, bringing out the relative importance of controls, boilers and chimneys in the satisfactory and efficient operation of oil burner installations.

Following Mr. Linn's main topic the meeting was opened to general discussion. Many questions were asked which indicated the keen interest in the subject. It was generally agreed among the 37 members and guests that the meeting proved very educational and should be beneficial in solving future oil heating problems.

Death of Charles D. Allan

At the February meeting of the Illinois Chapter announcement was made of the death of Charles Davis Allan on January 25, 1930, at his home, 4417 North Ashland Ave., Chicago.

Since 1914 Mr. Allan had practiced as a consulting engineer specializing in the mechanical equipment of buildings, designing and supervising the heating, ventilating, power, electrical and sanitary work.

Born in Fishkill-on-Hudson, N. Y., March 29, 1868, Mr. Allan received his engineering training at Cornell University, and completed the course in mechanical engineering.

Practically his entire life was devoted to mechanical fields and his first work in heating and ventilating was with the Buffalo Forge Co.

After six years he did erecting and testing of power plants for the American-Stoker Co., Erie, Pa., and then spent two years in engineering sales work for the American District Steam Co., North Tonawanda. Following this for a number of years he was with the Almiral & Co., New York City, and for two years prior to his entrance into the consulting field he was manager of the heating and power piping department for the General Fire Extinguisher Co., at Providence, R. I.

Mr. Allan was active in SOCIETY work, having served as President of the Illinois Chapter and on many of its Committees. He was also a member of the *A. S. M. E.*, the Chicago Rotary Club and other local organizations.

The many friends of Mr. Allan were saddened by the announcement of his death and the Illinois Chapter at its February meeting took appropriate action on the passing of their fellow member and conveyed their message of sympathy to his widow who survives.

Death of James Blair Walker

It is with deep sorrow that we learn of the death of James Blair Walker, past president of the Pittsburgh Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

While president of that Chapter, he gave constant evidence of his excellent business and executive ability and high character. His sound judgment and wise counsel, combined with his genial personality, strong character and innate integrity, contributed an inestimable service to the Pittsburgh Chapter during its formative years.

When the policies of this Chapter were first defined, he met the task of formulating and developing them with discernment and insight. His ability and influence will long be felt, both by those who honor him as a man and those who love him as a friend.

A resolution of bereavement was adopted by the Pittsburgh Chapter at a meeting held on February 10, 1930.

Proper Size of Coke for Domestic Heating Furnaces

Users of coke in domestic heating furnaces often have difficulty in deciding on the proper size to purchase in order to obtain the most satisfactory results, points out the United States Bureau of Mines, Department of Commerce. They also have little knowledge on how the adjustments of the dampers to give the best results are influenced by the size of coke used. The size purchased is usually decided by the advice given by the coal dealer. In districts where the sale of coke is well organized such advice will be based on the accumulated experience with the sizes available, but the relative results obtained by any one user when trying different sizes will depend on the draft produced by the chimney, the possibility of controlling the draft by the dampers, and on whether the size of the furnace is sufficiently large to heat the house without forcing the fire too much.

Extensive studies of the effects of the size of the coke have been made at the Pittsburgh Experiment Station of the Bureau of Mines. The settings of the dampers to obtain equally good results with different sizes were determined as well as the limiting size, larger than which it would be impossible to obtain such equally good results when using coke in an average size domestic furnace. It was found that the fire would go out at the low rates of burning required in domestic heating if the coke consisted entirely of pieces larger than those lying between a 2 and a 3-in. screen.

Comparing a small size coke (½ to 1 in.) with a large (2 to 3 in.), the advantages and disadvantages can be summed up as follows:

Advantages. With the small coke the favorable points are: (1) It is easier to start the fire; (2) it is less likely to go out; (3) there is more range of damper adjustment and thus the furnace will be more easily controlled; and (4) firings will not have to be quite as frequent.

Disadvantages. The poor qualities with small coke are: (1) A much greater draft is required; (2) a hotter fire can be obtained and thus worse clinkers may be formed; and (3) the efficiency will be lower, both because the carbon monoxide will be higher due to poor mixing of the gases and air, and because the exit temperature of the gases will be higher due to there being more combustion above the fuel bed and a higher initial temperature of the gases entering the passes.

This comparison shows that a small coke makes a fuel bed which burns more easily than the large size, but that the chimney must be capable of supplying sufficient draft or it will not be possible to obtain the required heat. On the other hand, the fact that it is possible to obtain a hotter fire with a small coke increases the chance of bad clinkers being formed. A small size becomes more desirable in mild weather and the large is less liable to give trouble at high than it will at low rates of burning.

It can be believed that the best average satisfaction would be obtained with a medium size of about 1½ in.

A complete report of these tests will soon be issued by the Bureau of Mines.

Mine Ventilation

A paper entitled "Fan Selection for Metal Mine Ventilation" was presented before the Mine Ventilation Committee at the Annual Meeting of the *American Institute of Mining and Metallurgical Engineers,* February, 1930, by N. L. Alison, Denver, Colo.

The Oil-Elec-Tric Engrg. Corp., recently formed at Minneapolis, took over the manufacture and sale of Oil-elec-tric burners, formerly assembled in the plant of the Campbell Machine Co., Minneapolis. C. G. Sigwald is chief engineer for this new corporation, and A. H. Coffin is in charge of sales and advertising.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 27 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1930, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ADAMS, HAROLD EUGENE, Chief Engineer, Nash Engrg. Co., So. Norwalk, Conn.	I. C. Jennings A. D. Harvey	R. H. Carpenter Russell Donnelly
BONDY, WINFIELD S., Clarage Fan Co., New York, N. Y. (*Advancement*)	C. S. Hoffman D. D. Kimball	William Kreitner C. H. Quirk
CALDWELL, ARTHUR C., Estimator & Engr., P. Gormly, Contrs., Philadelphia, Pa.	H. G. Black L. C. Davidson	A. C. Edgar R. F. Hunger
CAREY, PAUL C., Runyon & Carey, Newark, N. J.	Homer Addams J. I. Lyle	Perry West W. S. Koithan
CASE, WALTER GEORGE, Tech. Mgr., National Radiator Co., Ltd., Hull, England.	H. S. Downe Bernhard Goerg	H. G. Ullman O. O. Oaks
CONLEY, WALTER J., Ames Iron Works, Philadelphia, Pa.	H. G. Black J. M. Sewell	J. H. Galligan M. F. Blankin
FOULDS, SAMUEL T. N., The Trane Co., Boston, Mass.	R. F. Gifford I. S. Dane	W. T. Jones N. J. H. Shaw
GORDON, ROBERT H., American Carbonic Mchy. Co., Detroit, Mich.	G. H. Giguere H. C. Kappler	W. A. Blessed H. M. Stephen
HADJISKY, JOSEPH N., Albert Kahn, Inc., Detroit, Mich.	G. H. Giguere Dermid McLean	H. C. Kappler L. L. Smith
HAMMOND, MARTIN JOSEPH, Manager, American Heating Co., Long Branch, N. J.	H. G. Black J. M. Sewell	H. B. Gombers J. H. Galligan
HANBURGER, FRED W., Instr. in H. & V. & Plbg., College of the City of New York, New York, N. Y.	T. N. Thomson A. S. Armagnac	C. W. Presdee H. P. Waechter
HAYDEN, CARL F., Barber-Colman Co., Rockford, Ill.	D. J. Stewart G. L. Larson	H. G. Thomas H. R. Linn
HUSTOEL, ARNOLD MALVIN, Baltimore & Ohio Railroad, Chicago, Ill.	C. W. Doll (*Non-member*) B. A. Broom	H. D. Thompson (*Non-member*)
JACKES, HERMAN D., Sales Mgr., Aerofin Corp., Newark, N. J. (*Reinstatement*)	G. H. Giguere G. H. Cummins	H. M. Stephen E. H. Clark
JOHNSON, CARL ERIC, Buffalo Forge Co., Philadelphia, Pa.	L. C. Davidson R. F. Hunger	H. G. Black A. C. Edgar
KOZU, TAMIICHRO, Chief Engr., Daiwa Kogyo Co., Ltd., Kyobashi, Tokio, Japan.	Shigeyuki Kitaura Kunisuke Sekido	Kanjiro Yamasaki
MANNING, WALTER M., Heating Engr., Wigman Co., Sioux City, Iowa.	M. J. Orr W. V. Hagan	
MILLER, BRUCE RINKER, Sales Engr., Federal Steam Specialties, Oklahoma City, Okla.	R. L. Patton R. G. Dolan	F. X. Loeffler T. W. Rae
MURPHY, WILLIAM W., Treas., W. W. Murphy Co., Springfield, Mass.	A. J. Purcell W. T. Jones	T. F. McCoy R. S. Franklin
MYERS, GEORGE W. F., Mgr., York Htg. & Vtg. Co., St. Louis, Mo. (*Advancement*)	Thornton Lewis H. P. Gant	J. A. Carey J. V. Cavileer
PANCOAST, HENRY B., Jr., Treas., Henry B. Pancoast Co., Philadelphia, Pa.	H. G. Black R. C. Bolsinger	L. C. Davidson A. C. Edgar
PFEIFFER, JOHN FREDERICK, E. Keeler Co., Williamsport, Pa.	R. D. McLain M. C. Gillett	Charles Stewart A. J. Nesbitt

PORTER, RICHMOND CLAY, Research Asst., Engrg. Dept., Univ. of Kentucky, Lexington, Ky.	L. S. O'Bannon F. Paul Anderson	J. J. Lyle Thornton Lewis
RIESMEYER, EDWARD HENRY, Jr., Foreman, Schaffer Heating Co., Pittsburgh, Pa.	C. W. Wheeler H. E. Digby	G. H. Strachan S. E. Dibble
SONNEBORN, CHARLES, Shaw-Perkins Mfg. Co., West Pittsburgh, Pa.	E. C. Evans H. L. Moore	G. S. McEllroy Thomas Chester
SPOFFORTH, WALTER, Chief Engr., Ancker Hospital, St. Paul, Minn.	E. F. Jones Albert Buenger	E. J. Burns Webster Tallmadge
SULLIVAN, TIM J., Manager, Sullivan Valve & Engrg. Co., Butte, Mont.	R. L. Nelson R. H. Nelson	H. W. Nelson G. E. Otis
TEN BROOK, CHARLES SPAULDING, The Trane Co., La Crosse, Wis.	J. L. Blackshaw F. C. Winterer	A. J. Huch H. J. Sperzel
TICHENOR, LESLIE R., Jr., 1235 Salem Ave., Hillside, N. J.	E. C. Evans S. E. Dibble	H. L. Moore G. S. McEllroy
VAN COURT, WALTER GEARING, Htg. Engr. & Field Supt., Wm. H. Lutton Co., Jersey City, N. J.	F. E. W. Beebe Louis Sklenarik	A. F. Hinrichsen H. G. Issertell
WHITE, JAMES J., Vice-President, McArdle & Cooney, Inc, Philadelphia, Pa.	H. G. Black A. C. Edgar	M. F. Blankin L. C. Davidson

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

BRANDI, O. H., Manager, Carrier Lufttechnische Gesellschaft, Berlin, Germany.

CUCCI, VICTOR J., Kimball & Cucci, New York, N. Y.

DECKER, EMIL J., Automatic Coal Burner Corp., West New York, N. J.

DOLAN, HUGH P., Supervising Engr., Detroit Board of Education, Detroit, Mich.

FINNEY, GERALD J., Carrier Engineering Corp., Newark, N. J.

GIVELBER, SAMUEL HENRY, Secy. & Mech. Engr., The Kahn & Co., Cleveland, Ohio.

HEIMBERGER, OSCAR W., The Griscom-Russell Co., New York, N. Y.

HEROLD, CHARLES W., Herold Plbg. & Htg. Co., Kansas City, Mo. (*Advancement.*)

HOSKING, HOMER L., U. S. Radiator Corp., Detroit, Mich.

KEPPNER, HARRY W., Heating Contractor, 1310 South 56th Ave., Cicero, Ill.

MACDONALD, DONALD B., Br. Sales Mgr., C. A. Dunham Co., Kingston, Pa.

MCCOY, CLAUDE E., Genl. Supt., Eldorado Plbg. & Htg. Co., Little Rock, Ark.

RANK, DAVID J., Chief Engineer, Grant Bldg., Pittsburgh, Pa.

ROBINSON, MAYES R., Sales Engr., Bushnell Mchy. Co., Pittsburgh, Pa.

SCHUENGEL, GEORGE W., Sales Engr., N. Y. Blower Co., Chicago, Ill.

SNELLING, LOUIS R., Manager, National Radiator Corp., Buffalo, N. Y.

STRACHAN, GEORGE W., Chief Engr. of Design, C. A. Dunham Co., Pittsburgh, Pa.

WELLS, ERIC E., Civil Engr., Sarco Co., Inc., New York, N. Y.

ASSOCIATES

BERTHET, EDWARD E., Pittsburgh Br. Mgr., Skinner Bros. Mfg. Co., Inc., St. Louis, Mo.

COOPER, FREDERICK D., Excelso Products Corp., Buffalo, N. Y.

CROUSE, WAYNE W., Contr., Wayne Crouse Htg. & Plbg. Co., Pittsburgh, Pa.

DUNCAN, WILLIAM A., Service Engr., Dominion Oxygen Co., Ltd., Toronto, Ont., Canada.

DURKEE, MERRITT E., Heating & Piping Contractors New York City Assn., New York, N. Y.

JENNINGS, WARREN G., Res. Vice-Pres. & Br. Mgr., Minneapolis-Honeywell Regulator Co., Chicago, Ill.

KALLOCH, PARKER C., Jr., Pres., Fintube Radiator Co., Inc., Long Island City, N. Y.

PARKER, DUDLEY F., Eastern Mgr., National Regulator Co., New York, N. Y.

PENDLETON, MORRIS E., Dist. Mgr., Armstrong Cork & Insulation Co., Buffalo, N. Y.

PETERSON, STERLING D., Seattle Repr., Johnson Service Co., Milwaukee, Wis.

POHLE, KENNETH F., Dist. Mgr., W. F. Hirschmann Co., Inc., Boston, Mass.

RENTE, SIDNEY R., Sales Engr., Crane Co., Buffalo, N. Y.

ROARKE, HUGH B., Dist. Mgr., Detroit Stoker Co., Buffalo, N. Y.

SANDS, JOHN S., Westinghouse Elec. & Mfg. Co., Philadelphia, Pa.

SMITH, C. WARNER, Dist. Mgr., Ilg Electric Ventilating Co., Philadelphia, Pa.

TRAUGOTT, MORTIMER, Dist. Sales Mgr., Bryant Heater & Mfg. Co., Cleveland, Ohio.

WILSON, ANDREW W., Partner, Wilson Bros., Collingdale, Pa.

JUNIORS

ACKERMAN, REYNOLD H., Carrier Engrg. Corp., Newark, N. J.

ATKINSON, KENNETH B., Research Engr., York Htg. & Vtg. Corp., Bridgeport, Pa.

FOSTER, TILLMAN RICHARD, Carrier Engrg. Corp., Newark, N. J.

HERSH, FRANKLIN C., Carrier Engrg. Corp., Dallas, Texas.

JENS, JENS, Carrier Engrg. Corp., Newark, N. J.

JOHNSON, LESLIE O., H. Y. Keeler, Mech. Engr., Huntington, W. Va.

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MORRIS, JACK BRUCE, care of John E. Green, Detroit, Mich.

PARK, J. FRANK, Carrier Engrg. Corp., Newark, N. J.

STEWART, EARL ALBERT, Estimator & Engr., McVehil Plbg. Co., Washington, Pa.

WALTER, WILLIAM, York Htg. & Vtg. Corp., New York, N. Y.

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Relative Humidity

QUESTION:—Please explain relative humidity as it affects human beings and hygroscopic materials. (From a subscriber.)

Answer:—Consider a cubic foot of air, at room temperature, as if it were a cubic foot of ice. Put this cubic foot of air into storage and lower its temperature down to zero F. Endeavor to put moisture into this air and it is at once evident that no moisture can be added. All the moisture which the cubic foot of air will hold is 0.472 grains.

Raise the temperature of the cubic foot of air to 70 F. Enough moisture can now be added to bring the total content up to 8.069 grains. This is 16 times as much as the same cubic foot of air held at zero.

As a saturated condition existed at zero, the relative humidity was 100 per cent. Also, at 70 F, with 7.597 additional grains of moisture, the cubic foot of air was saturated and had a relative humidity of 100 per cent. In other words, under each temperature condition, the cubic foot of air was holding all the moisture it could possibly hold.

If at zero F this air held only 0.236 grains of moisture the relative humidity would have been only 50 per cent. Furthermore, at a temperature of 70 F, a moisture content of 4.0345 grains would have also represented 50 per cent.

It is, therefore, apparent from the examples given that the relative humidity of air is decreased when the air is warmed and increased when the air is cooled. Thus, if a cubic foot of saturated air is heated from zero to 70 F its relative humidity would have changed from 100 per cent to:

$$\frac{0.472}{8.069} = .58 \text{ or } \frac{58}{100} \text{ per cent}$$

Relative humidity, it is evident, takes into account not only the temperature of the air but also its absolute moisture content.

Nevertheless, when human beings or manufactured products are considered, the effect of the moisture in air is always noted in terms of relative humidity rather than absolute humidity or absolute moisture content.

The reason for this is that human beings, animals and birds, many manufactured products and many materials may be said to feel the dryness or dampness of the air due to hygroscopic conditions of the skin or of the exposed surfaces. This feeling of dryness or dampness is dependent almost entirely on the relative humidity and also, to a slight extent, on the air temperature. It does not depend at all on the absolute humidity.

Most people do not understand this fact because they do not know that all hygroscopic materials pick up or lose moisture due to the prevailing relative humidity. Thus humidity is frequently discussed without any consideration of whether the relative or the absolute humidity is meant.—*M. T.*

Heating · Piping and Air Conditioning EDITORIALS

Around the Corner

Newer and better ways of accomplishing a given purpose are the motive power behind all engineering effort. For many years air conditioning was the infant child of the heating and ventilating industry. Gradually, those few individuals and organizations interested in the control of atmospheric conditions felt the need for a broader knowledge on many phases of their work. Little by little the investment in research and development in air conditioning grew. As this investment became greater, per unit income, air conditioning forged ahead. Today air conditioning leads engineering in the exactness with which its efforts are applied and in the great benefits it bestows upon the human race. The infant of yesterday is the giant of tomorrow.

Right around the corner lies the future. Chemistry, physics and electricity will have an important share in the new developments of heating and ventilation. Not the least of these—electricity—has much to offer. Little vision is needed to foresee increased applications of electrical heating. Drying, in some instances, may be speeded up by the internal application of electrical energy. This use of radiant heat is growing by leaps and bounds. New and highly efficient controls, which are operated by photoelectric cells through electric relays using vacuum tubes, are coming into general use. Static electricity, in the near future, may be employed for the deposition of dust and moisture.

It is not improbable that relative humidity and effective temperature may, to a great extent, be maintained by electrical equipment. With the aid of hydration, a chemical process, air conditioning costs can be markedly reduced, in the low humidity field, so that wider applications can be made.

Chemistry is eliminating the need for air conditioning in some lines of endeavor. This elimination in one stage of a manufacturing process usually forces the adoption of air conditioning in allied processes. Thus, greater need goes hand in hand with elimination.

The physicist is showing us more efficient ways to use radiant heat and better methods by which equilibrium may be destroyed. His next job is the attainment of human comfort out-of-doors. The physician has already begun the use of air conditioning to cure some serious physical illnesses and is likely to find that it can also be used for certain mental cases.

Unbelievable as some of these prophecies may sound they are by no means out of the possibility range when one endeavors to look twenty years ahead.

Piping Economy

Simplicity and economy are the order of the day in all industrial operations and in piping systems particularly. In many piping systems, simplicity and economy are conspicuous because of their absence.

A certain complexity of piping is often necessary to insure continuity of operation. For this reason, duplication of piping is frequent, and complex piping systems are common.

A complex system should be so designed to permit of quick action when an emergency arises. If thought is not given to preparation for such emergencies, operation of the plant may be greatly hindered.

It is not uncommon that a plant or a part of a plant is shut down because of the closing or opening of the wrong valve. Often lack of identification of valves causes delays in repairing damage to equipment. Often pipe that is too crowded causes delay in making the repairs.

The economies that may be effected by the correct design and installation of the national annual production of between 1½ and 2 million tons of steel and wrought iron pipe is an item of immense proportions.

Air Cleanliness

Any machine will work better if kept clean. This is also true of the machinery of the human body. It is probably more necessary to keep the body clean inside than it is to keep it clean outside. In air conditioning, cleanliness of air is advocated for the purpose of keeping clean the lungs and nasal passages of human beings. In many factories it is also necessary for keeping the products of manufacture clean.

Unless he considers it carefully, the average man has no conception of the amount or kinds of particles in the air. Some few have a profound physiological effect and may be beneficial. The majority, however, are useless or harmful. The deleterious components of impure air generally may be classified as gases and dusts. Of the former, the gas most commonly found is CO_2. It is inert so far as its physiological reactions are concerned, but displaces the much needed air so that a person virtually may be drowned in carbon dioxide.

Dust is usually accompanied by a dry condition. There are the particles of carbon that float along in the smoke cloud. We enclose ourselves in boxes called homes, factories, skyscrapers, etc., in order to protect

ourselves from atmospheric elements and from thieves and in so doing we create a dry condition which favors dust. Carpets, draperies, floors and furniture give up particles by abrasion due to the movement of the people in the building.

Microscopic plant life such as that which is said to cause hay fever, bacteria, and germs and moulds and other fungus growth, float in the air either alone or riding on particles of dust. There are dusts which are the result of industrial processes in cement mills, tobacco factories, textile mills, etc.

There are several ways of preventing dust. One is by humidification of the air. When the air is moist, the dust particles will not break off and fly in the air as readily as when the air is dry. Also in moist air, the moisture, or what may be termed water dust, attaches itself to other dust particles and increases their weight and so they fall to the floor where they may be swept up and kept out of the lungs and nostrils of human beings.

Fumes and gaseous impurities are removed by replacing them with conditioned air. The practice is increasing of oxidizing certain fumes, gases and odors and dispelling them by the use of ozone or by the installation of equipment for ionizing the air.

Keeping in Touch

One of the most fundamental principles of good publishing is constant editorial contact with readers. Editors must know their readers—the work, the interests, the problems of those readers—if they are to supply the kind of articles that are wanted and needed. Readers can well appreciate what a problem it would be otherwise to prepare and present material month in and month out that would be live and close to the field's activities.

Heating, Piping and Air Conditioning from the beginning has taken very seriously its job of giving its thousands of readers the highest type of engineering and practical articles available on the subjects its name defines. Its staff has had pleasant and informative contacts with its readers, personally and by mail.

We were very much pleased to have many readers, in answering a recent questionnaire, take occasion to express their appreciation of our desire to know more about them in order to serve them better. We want them all to know that such contacts have been, are and will be prompted by a sincere desire to render them an improved editorial service and that we value highly their friendly, interested co-operation.

Sound

The American people are becoming increasingly sound conscious. Big hotels, hospitals, apartments and other buildings are being constructed in which a great deal of attention is being paid to sound prevention, and sound proofness of buildings is being advertised as a means of attracting patrons.

Practically all of the moving parts of heating, ventilating and air conditioning equipment cause noise. The ducts which carry heated or conditioned air are ideal for conducting sound from machinery and other noises from one room to another or from one floor to another. Many of the ducts function like a speaking tube frequently, with very uncanny and astonishing results.

It is predicted that the heating and air conditioning engineer will be compelled more and more to consider this new science of the prevention of sound transmission. He who specifies the equipment which makes or conducts noise will be held responsible for its prevention or elimination.

About the first question which arises in the providing of some new comfort or convenience is "how much?" "How much heat?" "How much humidity?" "How great an air velocity?" etc. The time has arrived when the heating and air conditioning engineer must differentiate in quantities of sound, for sound also is measured just as the various other influences which affect heating and air conditioning are measured, and it is predicted that the heating and air conditioning engineer shortly will become acquainted with the "decibel" which is the unit of measurement of sound—a term which has not yet found its way into the dictionary.

It is not uncommon these days to hear scientists speak familiarly about one-millionth of an inch and one millionth of a second so it may not be surprising that a decibel, explained in popular terms, is the amount of sound producing energy which equals the weight on the ear drum of a section of human hair of a length one-third of its diameter. A decibel is one-tenth of a bel. The energy required to produce sound goes in logarithmic progression. That is to say, the energy required is increased ten times for each unit of increase in loudness of sound.

The instrument used to measure sound is the audiometer. In the audiometer, a rheostat produces a buzzing sound. The loudness of the sound in the room or in the street is determined by the loudness of the buzzing sound that is required to mask the noise that is to be measured. It requires a certain electrical energy to produce the buzzing sound. The sound itself is not measured, but measurement is taken of the electrical energy required to produce the masking sound.

On the audiometer, decibels or units of energy are changed into terms of sensibility to loudness by divisions like those on a thermometer, which divisions range between no audible sound and sound that is so intense that it goes over the line from the sense of hearing to the sense of feeling.

If prevention of sound transmission by ducts and heating and air conditioning is to be the work of the air conditioning engineer, as many engineers predict, study of the articles on this subject which are to appear in Heating, Piping and Air Conditioning will be in order.

PRACTICAL PIPING PROBLEMS

Relocating Steam Supply Valves for Pipe Coils

Piping for Ammonia -- Threads and Joints

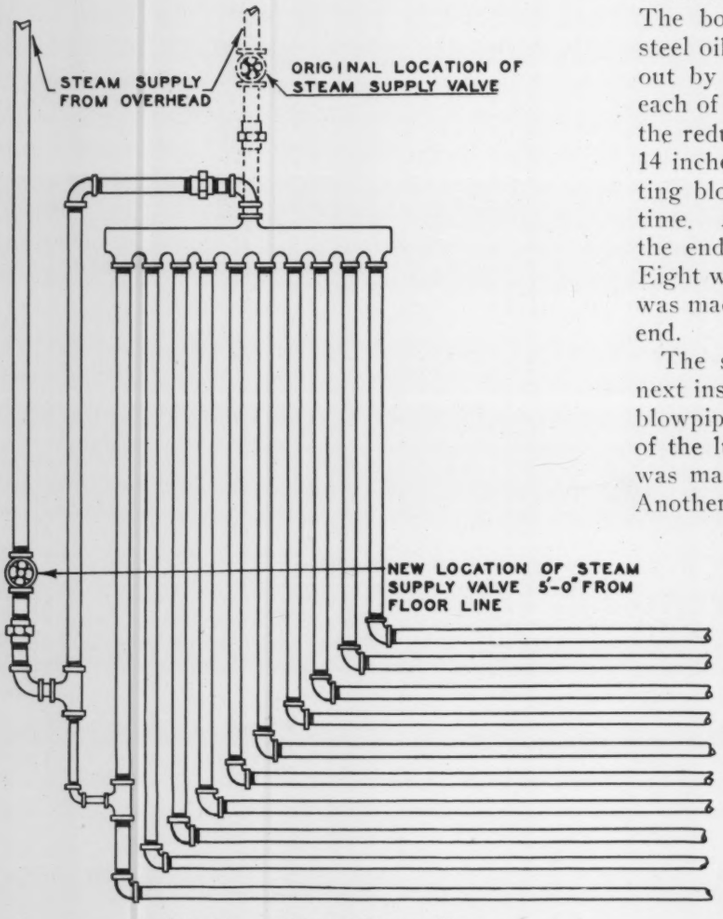
by **W. H. WILSON**

Location of Steam Supply Valve for High Steam Pipe Coils

The following method of relocating steam supply valves for steam heating pipe coils was used to advantage in a wood-working shop, where it was desired to have the valve in a convenient place to turn on or shut off the heat.

The pipe coils were of the miter type, of some length, located on the side walls with an overhead steam feed, low pressure steam being used. The vertical pipes to the supply manifold were, on account of the length of the horizontal pipes in the coil, long as compared with the usual type. The steam supply valve was necessarily at some distance from the floor when the supply was connected direct to the supply manifold.

In order to open or close this valve, a short ladder was kept on hand to be used by the employes. The result was that the valves were not used as frequently as they should have been, in order to control the heat. Doors and windows were left open in mild weather, instead of the ladder being hunted in order to close the valves.


STEAM SUPPLY FROM OVERHEAD
ORIGINAL LOCATION OF STEAM SUPPLY VALVE
NEW LOCATION OF STEAM SUPPLY VALVE 5'-0" FROM FLOOR LINE

LOCATION OF STEAM SUPPLY VALVE

The arrangement shown was used to place the supply valve in a position convenient for manual control. The steam supply was drained into the pipe coil, in order that the low pressure steam would not be trapped with condensation and seal the supply. This pipe can be connected to the outside vertical pipe of the coil, or if the manifold is tapped left-hand, a tee can be used in place of the elbow on the lower end of the vertical pipe and the drain pipe connected to the end of this tee.

Welded Oil Feeding System

There arose in an oil refinery in the Southwest an immediate demand for a lubricator to supply a group of five steam pumps, so one was made by the welding process in order to take care of the emergency.

Fig. 1 shows the design of the lubricator. It was constructed from standard materials, so there was no need to have special castings or parts fabricated. The body was made from a section of 15½-inch steel oil well casing, five feet long. After being laid out by means of templets, ten cuts were made in each of the two ends of the section or casing to form the reduced conical end. The depth of the cuts was 14 inches and all were made with the acetylene cutting blowpipe, and beveled for welding at the same time. After heating with the acetylene blowpipe, the ends were swedged in and the welding began. Eight welds were made on each end, and a final weld was made to fasten in place a 2-inch coupling at each end.

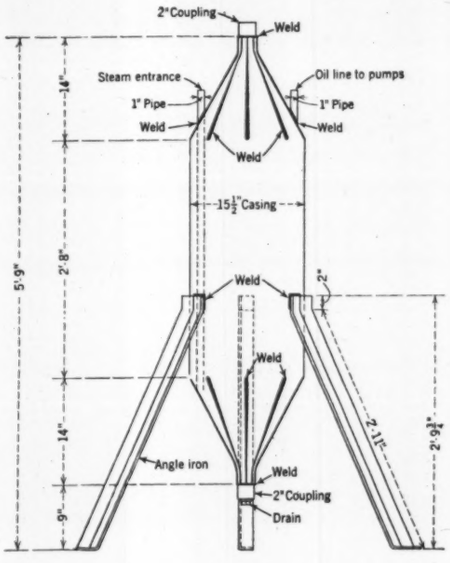
The steam feed pipe, one inch in diameter, was next inserted through a hole cut out with the cutting blowpipe. This pipe reached nearly to the bottom of the lubricator, as can be seen in Fig. 1. A weld was made to join the pipe at the point of entrance. Another piece of 1-inch pipe was inserted on the other side of the lubricator through a hole cut with the blowpipe, and was welded in place. This pipe, the outlet to the line through which the oil is forced to the pumps, was also welded at the top of the lubricator.

Three pieces of angle iron welded to the sides act as a supporting framework for the lubricator. A 2-inch plug was inserted in the couplings at top and bottom, the upper opening being used to fill the tank and the lower for a drain to draw off sediment and water.

The operation of the lubricator is briefly as follows: Steam enters through the 1-inch inlet pipe and condenses, the water going to the bottom of the lubricator and forcing the

oil up and out of the outlet pipe at the top of the lubricator and, from there, through the line to the pumps.

The making of this all-welded lubricator took one man's time for a day. Thus, the work of the welder



WELDED OIL FEEDER

in this emergency saved the refinery a possible shutdown due to improper functioning of the pumps under inadequate oiling.

Threads for Ammonia Piping

American threads are tapered and dies carefully made standardized to match the fitting taps. The practice in England has been to use parallel threads, which are much harder to handle. I have known threads on large pipes to be cut tapered on a lathe, which is an expensive method and if the taper is not right it is apt to cause great difficulty, so it is best to know what dies are being used.

After threads are properly cut and fittings are selected with good threads, don't throw threaded parts down in a pile of dirt. If this is done the dirt must be carefully cleaned off. For ammonia, the grease and dirt should be cut out with gasoline.

This takes quite a little care, but if you want to make sure that the thread is tight when put together it is the only way to insure it. Many pipe fitters do not take that trouble.

See that there is not a seam in the pipe which is carried down through the threads in a groove which will cause a leak. If pipes have seams, they should be rejected or cut off. Pipe should also have the full diameter and the burrs should be taken off inside. A usual way with some pipe fitters is to plaster the fitting in red lead or something, then screw in the pipe so as to make a neat appearing job on the outside. However, this squeezes quite a lot of stuff on the inside, which may make some trouble. Since the time taken to clean a thread is very small compared to the time of measuring and threading and screwing, and the trouble of fixing a leak is very great, it is best to use great care to see that the threads match and are clean. This applies to all pipe work.

Ammonia will seep through red and white lead, or almost any cement, except litharge. Litharge is very good, but it should be mixed fresh with glycerine and generally just enough for each joint, which is a little trouble.

Before applying any litharge, the pipe should be entered about three or four threads by hand to make sure that it will go in about the proper depth and the thread should never be soldered outside; that is after screwing home, there should be about two threads visible on the outside. These threads are never cut the full depth and after the pipe is screwed in until they butt, it means that the main inside threads do not bed properly. Pipe is very solid and will shake loose and leak.

It is possible to get pipes tight, metal to metal, without litharge by using oil as a lubricant. However, the pipe lengths must be screwed very tight and generally litharge is a little safer and makes a good job, although it sticks very hard, making it difficult to take joints apart.

Soldering Joints

If a pipe is tinned in solder and screwed into a fitting tightly it will make a good job and if screwed in very tight into very heavy fittings, it is probably as permanent a job as can be had. It should not be heated afterwards, or the inside solder will be melted and the pipes will become loose.

Hence, if sweating is needed or desired on the outside, the pipes should be screwed in tight so they will set solid, even when heated, to run the solder into the groove. All fittings don't take solder easily, and the so-called sweated joint has to be done very carefully or the solder may be porous and not stick and hence be likely to leak, so a sweated joint is not necessarily better than any other. Having solder around the outside of the thread protects it from rust, although litharge will also protect the thread if properly applied.

When soldering is done in a shop, fittings can be heated in a bath of solder, pipes hoisted vertically and screwed in and then moved over while the fittings and pipe are hot. The joint may be sweated, using acid and flux while the fittings are hot.

Sealing Ammonia Pipe Work

Most gases are likely to penetrate metal and joints much more easily than liquids, some gases more than others.

Ammonia is much worse than air in this respect, probably because air is likely to have moisture in it, and after passing through pores, it has a slight rusting effect which helps seal the openings.

This sealing can be done in the case of ammonia, to a reasonable extent, by the use of sal-ammoniac, which can be forced into the pores of metal where it rusts up. Sometimes this method can be used to make a good permanent job.—*H. T.*

A "Directory of Research on Heat Transmission in Educational Institutions in the United States and Canada" has been published by the National Research Council through its Committee on Heat Transmission. This directory is devoted to the research at thirty-six colleges and universities, one page of the publication being allotted each institution.